The

ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

OL. III. No. 11 NOVEMBER, 1942

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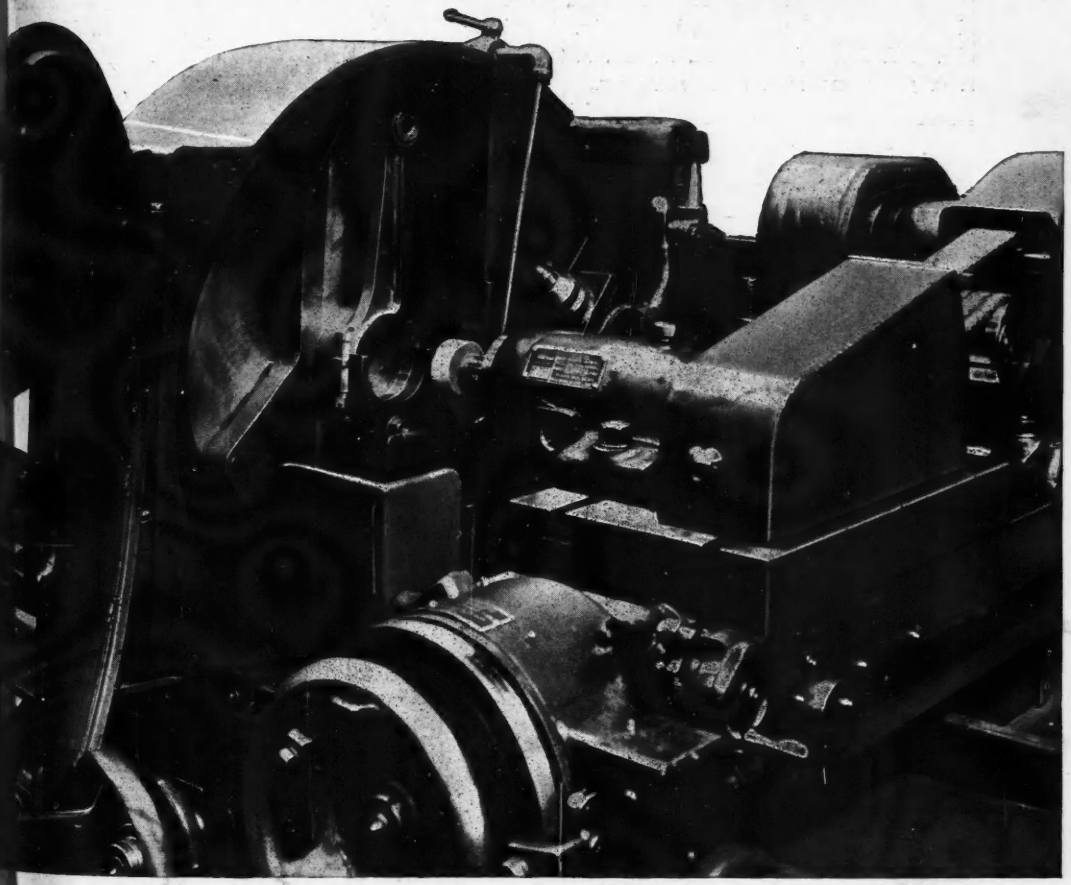

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The radiograph reveals

what destructive examination confirms

This is an instance of the value of radiographic examination. The blow-hole in the aluminium casting is revealed in the radiograph, yet the casting remains undamaged and fit for use if the flaw is judged not to be serious. In this case, however, the evidence of the radiograph has been confirmed by sectioning the casting to expose the flaw—which is clearly seen in the photograph of the sectioned casting. Thus in the routine testing of castings radiography ensures flaw detection, but leaves the casting intact. 'IndustreX' D, 'IndustreX' S and 'CrystalleX' are X-ray films specially produced for such industrial radiographic purposes as the examination of castings, welds, and other engineering produce.

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TESTING OF ALUMINIUM CASTINGS.

By PROFESSOR DR. A. VON ZEERLEDER. (From *Die Giesserei*, Vol. 29, No. 1, January 9th, 1942, pp. 7-10).

THE object of testing materials is to collect information of their properties. The test must, therefore, correspond to the requirements of the application for which the material is to be used.

Whilst in sheet metal the strength is generally the same in all directions, in strip, as a result of the rolling in one definite direction, different properties are obtained in the longitudinal and transverse directions. In forgings there are variations in properties with varying section because of varying degrees of working. In heat-treated alloys, the strength is less with thicker sections due to less effective quenching. Even greater variations in properties can be observed in castings, depending on the casting conditions. With aluminium castings, the grain size, and hence the mechanical properties, are greatly influenced by the rate of cooling; the finer the grain, the greater is the strength.

Since the rate of solidification governs the grain size, the cross-sectional area of the casting and the conductivity of mould are important factors. Metal moulds have better heat conductivity than sand moulds, so that in general die castings have better mechanical properties than sand castings. This is shown in Table I. The

TABLE I.

Comparison of the strength of sand and die-castings.

Class	Alloy	Mean tensile strength in kg/mm²		Strength of sand-casting expressed as a % of the die-casting
		Sand casting	Die-casting	
Al-Zn-Cu	German Alloy	15.5	18	86
Al-Cu	U.S.A. Alloy	14.0	17.5	80
Al-Si	Silumin α	17.5	20.5	85
Al-Si-Mg	Silumin β (as cast) ..	18.5	24.0	77
Al-Si-Mg	Silumn γ (fully hardened)	27.0	29.0	93
Al-Cu-Ni	Y alloy (as cast) ..	17.5	23.0	76
Al-Cu-Ni	Y alloy (fully hardened)	25.0	27.0	93

effect of the cross-sectional area of the casting upon strength is shown in Fig. 1. It can be seen that the relationship, thickness-strength, does not follow a straight line. The percentage decrease of strength with increasing thickness is the least for Al-Si-Mg alloys when fully hardened, and the highest for the Al-Cu alloys, with or without addition of Mg.

The pouring temperature has other effects in addition to influencing the rate of solidification; increasing the pouring temperature does not always give reduced strength, but actually increases the strength in certain alloys. This is mainly due to the beneficial effect of high pouring temperature upon the formation of blow holes.

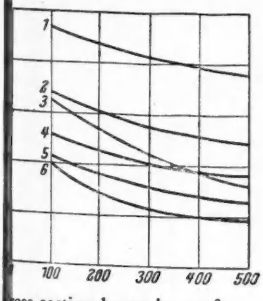


Fig. 1

Influence of cross-sectional area on the strength of sand-castings.

1 = GAl-Si-Mg fully hardened,
2 = GAl-Mg-Si incompletely hardened
3 = GAl-Cu-(Mg) incompletely hardened
4 = GAl-Si
5 = GAl-Zn-Cu
6 = GAl-Cu (8%)

A survey of the strength of castings shows that even for very simple sections the variation of strength is large, depending upon the conditions of castings. Fig. 2 shows the strength at various points of a cast bush made of Anticorodal (2.5% Si, 0.5% Mg, 0.5% Mn) for various conditions of casting.

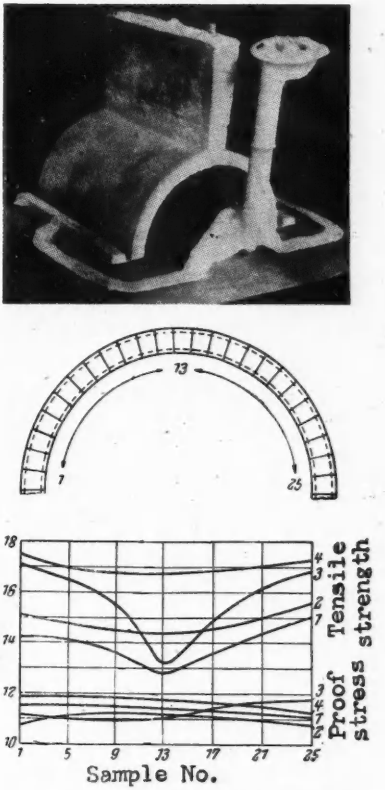


Fig. 2. Influence of the method of casting on the mechanical properties.

Curve 1 was obtained using a low runner on one side of the bush and equally low risers on the opposite side and at the centre. In curve 2 the runner and risers were arranged higher while in curves 3 and 4 the risers were as high as in curve 2; in curve 3 the centre riser covered only one third of the casting, whilst in curve 4 it covered the entire length.

As the variations are so great even for the simple casting described above, it is not surprising that for a more complicated casting, such as, for instance, a heavy lorry wheel, cast from Aloufont (2 Si, 4 Cu, 0.2 Mg, 0.6 Mn, 0.15 Ti) the difference becomes even greater.

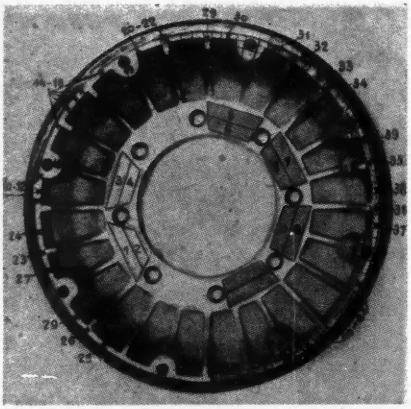


Fig. 3. Heavy lorry wheel cast from Aloufont.

In this case proof stress figures of 14.8 to 23.2, tensile strengths of 19.1 to 26.5 kg/mm² and elongations of 1 to 3.3 % were observed. It should be noted that the most favourable conditions of casting had been predetermined and employed.

The few examples given above serve to throw some light on the possibility of testing materials by using cast test bars.

If exact data are required concerning the strength of a casting, they can only be obtained by taking a test piece from the actual casting and thereby destroying it. Even so, the strength varies with individual castings, depending not only on the temperature of pouring but in the case of sand castings, also on the water content and the tightness of packing of the sand. Only very seldom can a casting be cut up into pieces, and therefore, the determination of the mechanical properties is reduced to a test of the quality of the metal used.

There are widely differing opinions as to whether it is permissible to cast the test pieces separately or whether to incorporate them in the casting. Considering the great differences in strength found in any one casting, even the cast-in-one test pieces would only represent the strength at one particular point. Fig. 4 shows how test pieces or blocks from which test pieces are subsequently machined may be incorporated in a casting.

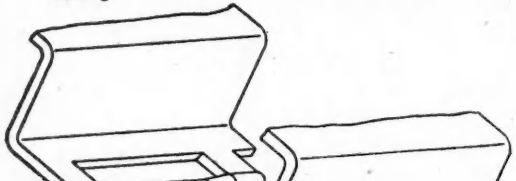
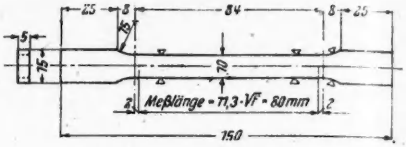


Fig. 4a. Cast-in-one test sample

Fig. 4b. Metal block (for machining).

If a cast-in-one specimen is required, this is machined from a plain block, 5 mm. thick as shown in Fig. 4c, its final dimensions being 5 × 10 mm. The 10 mm. faces of the specimen remain unmachined. If there is



(Messlänge = Gauge length).

Fig. 4c. Test sample machined from block.

insufficient room, a gauge length of $\frac{11.3}{2}\sqrt{A} = 40$ mm. may be used, thus reducing the overall length from 150 to 110 mm. There are considerable difficulties in the casting of test specimens as shown in Fig. 4a, because casting defects are known to be more frequent at such places, and for this reason a large number of test samples must be obtained from each casting in order to give sufficient information. This increases the cost of the mould considerably. In view of the fact that any one cast-in-one test piece does not represent the true strength of the casting, the separately cast specimen is usually preferred. The only drawback of this method is the danger of mixing up the test pieces; with suitable marking and reasonable care, however, this can be avoided.

If we realise that, in the first place, the test piece has to indicate the properties of the liquid metal at the time of casting, then separately cast test bars are to be preferred on account of their greater uniformity and reduced numbers. The mould used for the test piece must be similar to that used for the casting, i.e., for sand castings similar sand, and for die castings, chill-moulds of similar temperature must be used. This measure is particularly important because castings of widely different properties are obtained when the mould or the rate of solidification is altered.

Fig. 5 shows the most suitable shape for test bars cast in sand moulds. As castings are frequently used without being machined, the specimen should be tested in a similar condition; time and labour is thus saved. It is important to ensure that as good castings are obtained as test pieces. Great care must be taken in the case of specimen made by die casting to allow for shrinkage, and early opening of the mould is advisable.

Summing up it can be said that the test piece, whether cast-in-one or separately cast, gives only an indication of the strength of the casting. The cast-in-one specimen has no advantages in this respect, being more costly and even influencing the strength of the main casting itself; thus separately cast test pieces are to be preferred.

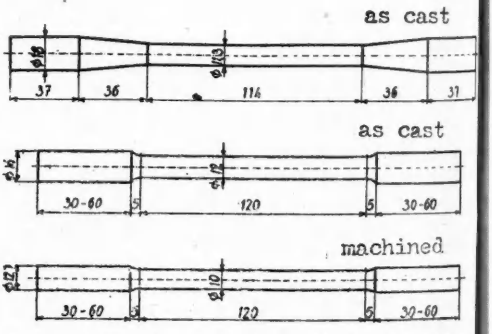


Fig. 5. Dimensions of test sample for aluminium castings.

NEW 2000-B.H.P. DIESEL-ELECTRIC ROAD LOCOMOTIVES

By FRANK GUILLOT. (From *General Electric Review*, Vol. 45, No. 6, June, 1942, pp. 326-30).

THE first diesel-electric locomotive for the New Haven system was placed in switching service in 1932. The success of this type of locomotive is indicated by the fact that today the New York, New Haven and Hartford Railroad Company is operating approximately 60 diesel-electric locomotives of various sizes, including twenty 2000 H.P. diesel-electric locomotives recently built by the American Locomotive Company. Two of these units, when coupled together as in Fig. 1 handle as many as 18 passenger cars, and freight trains, up to 4500 tons, on regular schedules between Boston and New Haven. Single units are used on trains up to half of these sizes. The new locomotives are built for both passenger and freight service with 80 m.p.h. top speed. They weigh 165 tons with a weight of 230,500 lb. on the driving axles.

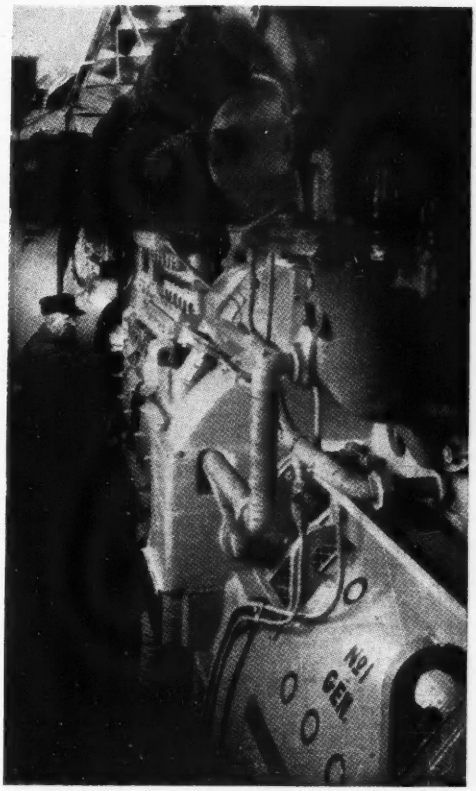


Fig. 2

Power Plant.

Each locomotive is equipped with two 1000-B.H.P. 6-cylinder 750-r.p.m. super-charged American Locomotive Company diesel engines, each direct connected to a General Electric D-C generator providing power for two traction motors. The engines and generators are shown in Fig. 2, while the control arrangement is illustrated in Fig. 3.

General Dimensions and Weights of 2000-B.H.P. units.

Length (over-all—over pilot)	74 ft. 8 in.
Width (maximum)	10 ft. 6 in.
Height (maximum)	14 ft. 4 in.
Weight on drivers (4 axles)	230,500 lb.
Weight on idlers (2 axles)	115,500 lb.
Weight—total locomotive (working order)	346,000 lb.
Wheel base—each truck (rigid)	15 ft. 4 in.
Wheel base—total locomotive	58 ft. 4 in.
Wheels—drivers (4 pairs) diam.	40 in.
Wheels—idlers (2 pairs) diam.	40 in.
Maximum speed restriction	80 m.p.h.
Starting tractive effort (at 23% adhesion)	53,000 lb.

Electrical Equipment.

Each traction generator set consists of a main generator, an auxiliary generator and a belt driven split-pole exciter, assembled into a complete unit. The main generator is direct connected to the engine through

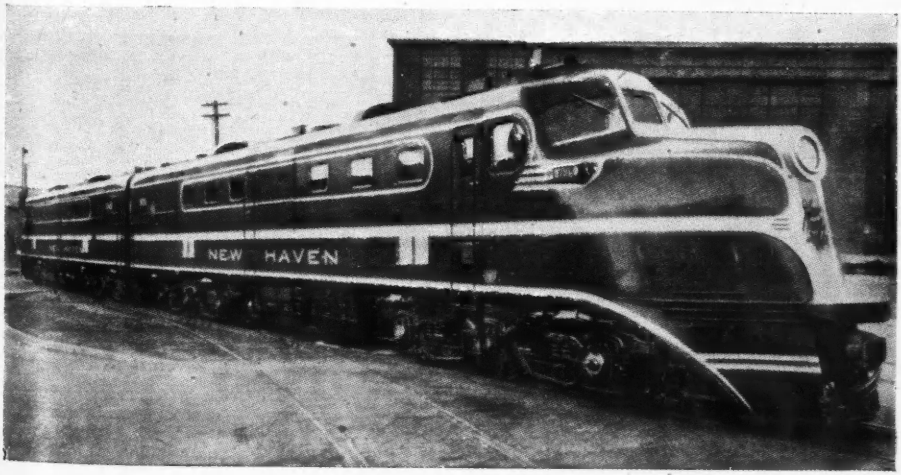


Fig. 1

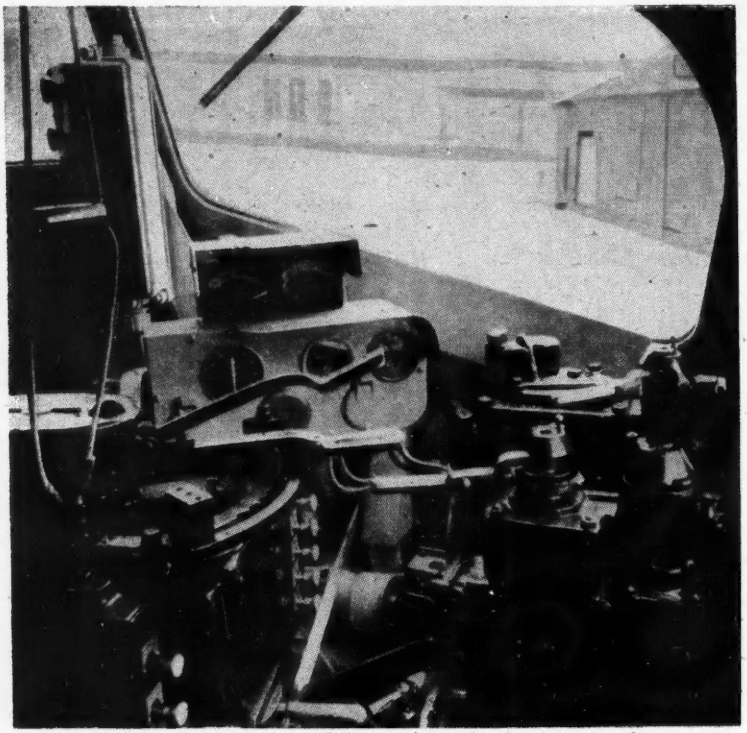


Fig. 3. Interior view of locomotive cab showing controls.

a rigid coupling supported by the main engine bearing. A single main-generator self-aligning bearing supports the commutator end of the armature. The main generator shaft is extended for driving and supporting the auxiliary generator armature.

Cooling air is drawn through longitudinal armature ventilating ducts as well as over the armature field surface from the commutator end toward the engine end.

A split pole exciter supplies the main field excitation and governs the output characteristic of the generator.

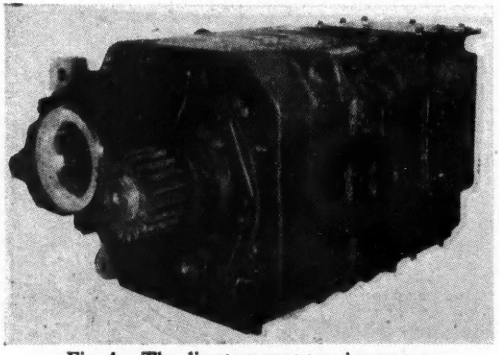


Fig. 4. The direct-current traction motor.

A separate starting field is built in for operating the generator as a motor for starting the engine. A self excited auxiliary generator supplies power for charging the battery, for all the lighting and for operating the control. Voltage is controlled by a relay, which holds the auxiliary voltage constant over a wide range of engine speed. The split pole exciter is mounted on top of the auxiliary generator frame and belt driven from the main generator main shaft. It supplies excitation for the main generator field. The exciter field poles are split into two sections and are excited by shunt and differential windings. The shunt winding, excited through the control from the storage battery, and the differential winding connected in series with the main traction motor circuit together with the engine speed and proper control, result in the main generator delivering its designed output.

Traction Motor.

One of the four pole D-C commutating pole motors is shown in Fig. 4.

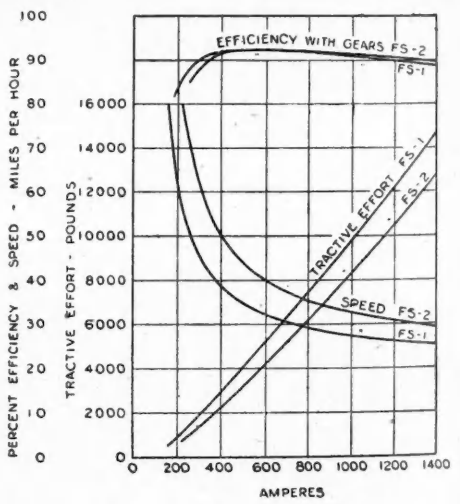


Fig. 5. The 600-volt railway motor characteristic curves for full-field (FS-1) and reduced field (FS-2) operation. (Gear : 64 teeth ; pinion : 19 teeth ; reduction : 3.367 ; wheel diameter : 40 in.).

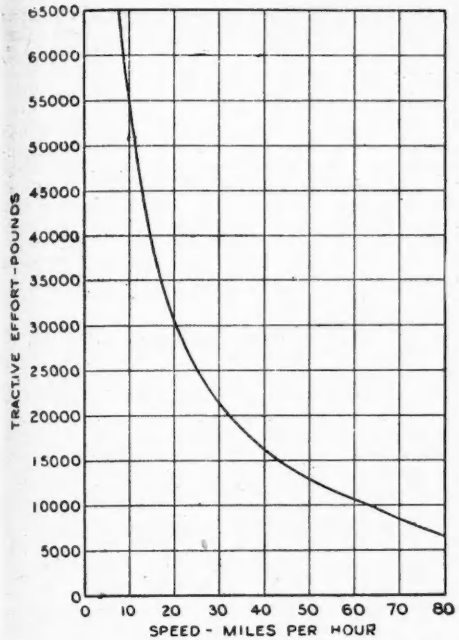


Fig. 6. Speed-tractive effort curve for 2000 H.P. locomotive.

They have a continuous rating of 700 amp. each and drive the axles through 64:19 spur gears. A total of 1700 cu. ft. of air is furnished to each motor from two belt driven blow units. These motors give the locomotives the speed-traction effort characteristic shown in Fig. 6.

Method of Control.

To meet all causes of horse power variation, which might arise, there has been designed a system of generator control, which automatically varies the engine load with the horse power output. This is accomplished by the combined speed-switch split-pole differential-exciter system of regulation as employed on Alco-G.E. road locomotives.

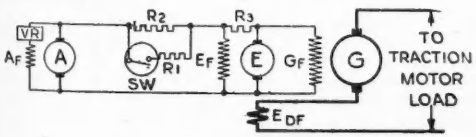


Fig. 7

Simplified connections for supervisory speed-switch split-pole differential-exciter system of generator regulation used on the locomotives.

G Main generator
A Auxiliary generator
E Split-pole differential exciter
SW Speed switch
VR Voltage regulator
R_1, R_2, R_3 : Resistances
G_F Main generator exciting field
E_F Exciter shunt field
E_{DF} Exciter differential field
A_F Auxiliary-generator exciting field

The barest essentials of the scheme of connections employed are shown in Fig. 7. A, G and SW are mounted on the same shaft, while the exciter E is belt driven from the shaft. The speed-switch SW is arranged and driven so, that the faster the engine turns over, the greater is the percentage of time during which the contacts are closed. Closed contacts of SW reduce the resistance in series with the exciter field, E_F and the value of E_F becomes a maximum.

There is a point at which the contacts in the speed-switch are open for 100 per cent. of the time. Under this condition the regulation is that of an ordinary split-pole exciter system having a characteristic corresponding to (a) in Fig. 9. When at a certain maximum speed the contacts would be closed continuously, the regulation is that of an ordinary split-pole exciter having a characteristic as shown by (b) in Fig. 8. However, this characteristic represents a horse-power output greater than that which the engine is capable of delivering, and is, consequently, never attained. With the engine putting out its rated horse-power the curve (c) will be obtained. That part AB of the top characteristic, which is less than full horse-power, is available because of the action of the speed switch. The additional full-load portion BC is obtained also. The supervisory speed-switch differential split-pole system not only takes care of variations in engine output capabilities, but also provides the full available horse-power over a wider range of generator loading.

In Fig. 8 curves (a'–), (b') and (c') represent voltage characteristics imposed by the differential winding of the split-pole exciter under different conditions of speed-switch operation, while curve (e) represents the voltage characteristic due to the cumulative winding of the exciter.

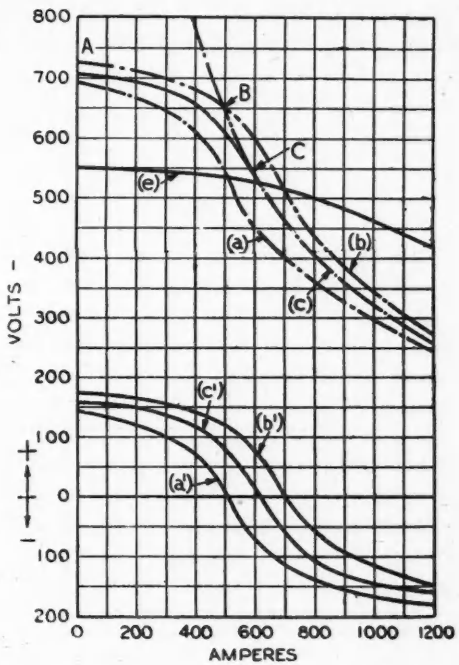


Fig. 8. Characteristic curves for speed switch and split-pole differential exciter system of generator regulation.

Curve (a) represents the voltage characteristic obtained when the speed switch is open 100 per cent. of the time. That part of curve (a') above the zero-excitation line, added to the corresponding load section of curve (e), results in the main generator characteristic as shown in the upper part of curve (a); while the section of curve (a') below the zero-excitation line, subtracted from the corresponding load section of curve (e) results in the lower part of the main generator curve (a).

Curve (b') represents the voltage characteristic imposed by the differential winding of the exciter when, at a certain maximum engine r.p.m., the speed-switch contacts are continuously closed. This curve combined with curve (e) as previously explained, results in the main generator characteristic curve (b) in Fig. 8.

Curve C, lying between curves (a) and (b) represents the rated horse-power main-generator characteristic curve and is the resultant combination of curves (c') and (e).

Control Connections.

The generators are not paralleled. Each unit supplies traction energy to two traction motors independently from the other generator. The traction motors of each power unit are connected so that they can be operated two in series, two in parallel full field, and two in parallel with one step of reduced field. The transition from series to parallel full field and to parallel reduced field is effected automatically under the control of a relay operating on generator voltage.

BENDING MOMENT AND ENERGY OF PLASTIC DEFORMATION.

By H. BIRKLE, VDI. (From *Die Werkzeugmaschine*, Vol. 46, No. 12, June, 1942, pp. 353-356.)

IF the process of bending and straightening of sheet results in permanent deformation, the stresses in the cross-section of the sheet exceed the yield point of the material. The relations of elastic straining cease to apply. The bending moment $M = W\,\sigma$ must be the lower below the value of the actual bending moment M_1, the greater the product $\sigma\,\epsilon$, that is, the more the yield stress σ_f is exceeded and the smaller the radius of curvature R becomes. (W = modulus of section, σ = stress, ϵ = strain.)

By denoting $\frac{M1}{M} = n$, then n represents the correction by which the bending moment $W.\sigma_f$ is to be multiplied in order to obtain the plastic bending of M_1.

For the bending of sheets of rectangular cross-section the bending moment in the fully plastic state is known from earlier investigations to be $M_1 = \frac{s^2 b}{4} \cdot \sigma_f$ (s = sheet thickness; b = width of sheet).

The laws connecting elastic and plastic deformation will be shown by means of an example using a rectangular sheet cross-section. The fully plastic state is reached when the material shows distinct yielding and the ration of $R/n \gtrless 100 : 1$ (Fig. 1).

In Fig. 1 :—

(a) the stress strain curve of a carbon steel with a yield stress of 30 kg/mm² (42,800 lb./in.²);

(b) the relation between strain and radius of curvature $\epsilon = \frac{\Delta l}{l} = \frac{a}{R}$;

(c) the stress distribution over the cross-section when the outermost fibre reached the yield stress, corresponding to a strain $\epsilon = \frac{\sigma_f}{E}$ or a radius of curvature $R_1 = \frac{a\,E}{\sigma_f}$

(d) the stress distribution over the cross-section in the almost fully plastic state;

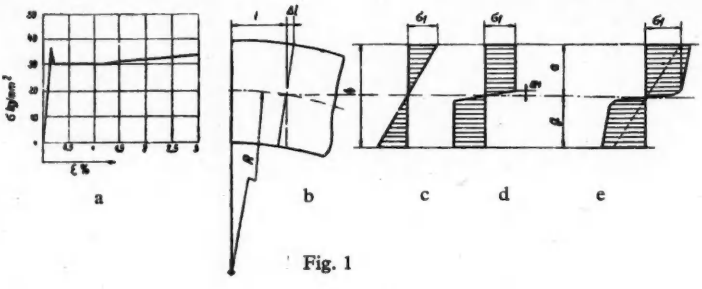


Fig. 1

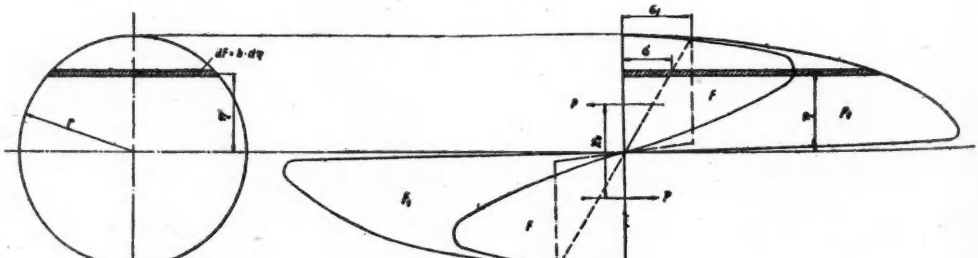


Fig. 2

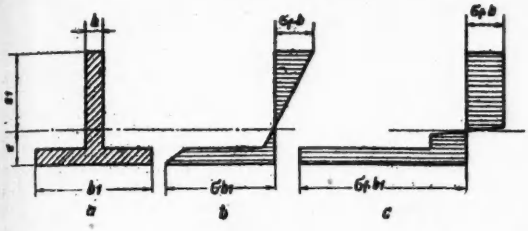


Fig. 3

(e) the stress distribution in an alloy-steel of high strength for a condition as in "d."

Fig. 2 illustrates the construction of a load diagram for section by taking strips of the area dF and multiplying each area by the corresponding stress σ. The area thus obtained is the tension—compression load-area F. For cross-sections of constant width b, these areas are similar to the stress areas so that the ration of the static moments $\left(\frac{S_1}{S}\right)$ of the stress area in Fig. 1c and Fig. 1d represent the coefficient $n = \frac{M_1}{M} = \frac{S_1}{S}$

According to Fig. 1c and 1d:

$n = \frac{S_1}{S} = \frac{\frac{1}{2}\sigma_f a_2^2}{\frac{1}{2}\sigma_f a} = 1.5$ for rectangular cross-sections.

In a general way:

$$S_1 = \Sigma\sigma_f . dF. \eta\,;\quad S = \Sigma\sigma_f dF. \eta = \frac{\sigma_f}{a}\Sigma dF. \eta^2,$$

by substituting $\sigma = \frac{\sigma_f \eta}{a}$

Thus, $S = \frac{\sigma_f}{a} . I = \sigma_f . W,$

and $\frac{S1}{S} = n = \frac{\sigma_f \Sigma dF. \eta}{\sigma_f W} = \frac{\Sigma dF. \eta}{W} = \frac{S_2}{W}$,

where Σ dF. η is the static moment of the cross-section, denoted by S_2.

For a circular cross-section:

$$n = \frac{S_2}{W} = \frac{4r . r^2 \pi}{3\pi} \Big/ \frac{\pi . 8 . r^3}{32} = \frac{16}{3\pi} = 1.7.$$

For cross-sections such as shown in Fig. 3a, which are unsymmetrical with respect to the flexural axis, the formula $n = \frac{S_2}{W}$ holds good too. Fig. 3b shows the tension and compression load area for the case when the extreme fibre reached the yield point, and Fig. 3c after the fully plastic state has been reached. Here, for W the value $\frac{I}{e_1}$ is to be substituted.

For the triangle, Fig. 3d, one obtains:

$$n = \frac{1}{\sigma W}[a . h_1^2 + h^2(a + 2b)] \sim 2.35.$$

Table I. contains values of n about two flexural axes for the most usual sections. For carbon steels the values of n are valid up to 3–3.5% straining at the outermost fibre. With increasing strain the σ values increase again, and n can be calculated from the stress diagram (e.g., Fig. 1e). Maximum strain is reached when the inner radius of curvature is about equal to the height of the cross-section. In practice, however, this is not realisable, except when heated to about 500°–600° C. In the calculation of M, the strength at 500°–600° C. is replacing σ_f.

TABLE I.

Profile	n	Profile	n
	1.6		1.5
	1.5		2
	2.35		
	1.7		
	1.3		
	1.9		
	2		1.5
	1.18		1.6
	1.12		1.5
	1.2		2
	1.35		1.6

Having calculated the plastic bending moment, the required bending energy A can now be estimated. If P is the bending force, $A = 2P . \epsilon 2R\pi$. From Fig. 2, $P = \frac{M1}{X_o}$, and $\epsilon = \frac{X_o}{2R}$. Thus, $A = \frac{2\pi M_1 \alpha^o}{360^o}$, and for the bending to a circular ring: $A = 2\pi M_1$.

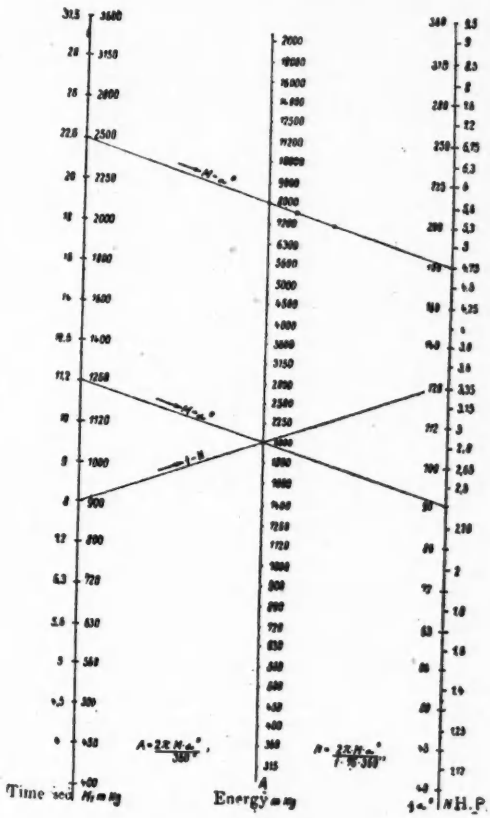


Fig. 4

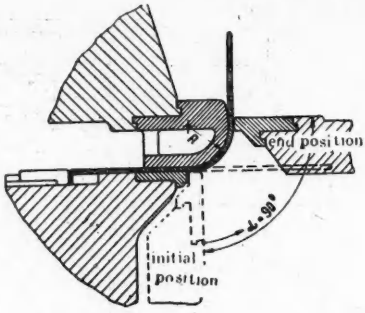


Fig. 5

The energy of bending is equal to the energy of developing a rolled-up strip of $2\pi R$ length by applying a tensile force P.

Herewith, P. $2\pi R = 2\pi M_1$, or $P = \frac{M_1}{R}$

The power required for bending is $N = \frac{P.v.}{75} = \frac{M_1 n}{716.2}$ H.P.,* where n is the r.p.m. of the shaft.

The nomogram in Fig. 4 shows the bending energy A, and the required power N. As an example, a sheet of 10 mm. thickness and 1,650 mm. width should be bent through 90° with a radius of curvature of R= 200 mm. The actual set for this operation is shown in Fig. 5.

*H.P. stands here for metric horse power.

The bending moment is :

$$M_1 = \frac{1.5.\ W.\ \sigma_f}{100} = = 1250 \text{ mkg.}$$

It takes 8 seconds to complete the bending operation. From the nomogram $N \cong 3.35$ H.P. join M_1 at the inner left-hand scale with α at the inner right-hand scale by a straight line and draw a line through the point where it intersects the middle scale and t = 8 sec. at the outer left-hand scale. The extension of this line indicates the value of N at the outer right-hand scale. The value of A can be read off directly. E.g., $M_1 = 2{,}500$ mkg. and $\alpha = 180°$ are connected by a straight line intersecting the middle scale at A=8,000 mkg.

The theoretically required power for bending N, obtained from Fig. 4 divided by N_t the actual power output of the motor, which can be measured, gives the efficiency $\eta = \frac{N}{N_t}$. η has values generally between 0.4 and 0.6.

The value a, shown in Fig. 1d, represents an error in the calculation of $M_1 = n.W.\ \sigma_f$. M_1 so determined is greater by the amount of this error as a fully plastic state has been assumed. However, calculation shows that for sections with a radius of curvature R = 15 h to 50 h, a_1 is only of the order of 0.02 h to 0.07 h, and the assumption of full plasticity appears to be justified.

The plastic bending moment M_1 is therefore the basis for calculation of plate bending or straightening machines.

If $R \geq 250$ h, a mean value for n should be used, which, taken as $1 + \frac{n}{2}$, is generally sufficient.

STRESSES AND DEFORMATION UNDER TORSION OF THIN-WALLED CYLINDERS WITH CIRCULAR CUT-OUTS.

Part I. by D. THOMA, Muenchen. Part II. by M. SCHILHANSL, Heidenheim/Brenz. Report of the J. M. VOITH ENGINEERING WORKS, Heidenheim/Brenz. (From *Luftfahrtforschung*, Vol. 19, 1942, No. 6, pp. 210-214.)

PART I.

STRESSES in a thin-walled tube without a cut-out subject to torsion can be imagined to be produced by rolling of a flat plate under pure shear stress to a cylinder, whereby the stresses remain unchanged at all points and the state of equilibrium is undisturbed.

If, however, a flat plate with circular cut-out and in pure shear at a greater distance from the cut-out is rolled to a cylinder, displacements normal to the wall will disturb the equilibrium at the proximity of the cut-out. This will cause bending stresses and deformation of the wall near the cut-out, and the question arises now what additional, normal, external, evenly distributed forces should be applied in order to bring the tube-wall back to cylindrical shape, eliminating of course the bending stresses. If the additional force required is known, it is possible to estimate the deformation and the actual bending stress existing after removing the additional force.

Fig. 1 shows the notation : a is the radius of the cut-out, R the radius of the cylinder, τ_∞ is the shear stress (infinitely) far from the cut-out. The developed surface of the cylinder is shown in Fig. 2. Another simplifying assumption is that a is very small compared to R, so that τ_∞ at the edges seems to be justified.

This state of stressing can be described by using the rigorous solution of Kirsch (Z.VDI., Vol. 42 (1898), p. 797), for a plate with a hole in it under direct load. By superimposing on the tensile stress, I, the compressive stress II, each of a magnitude equal to τ_∞, (Fig. 3), the following stresses in Polar co-ordinates are resulting :

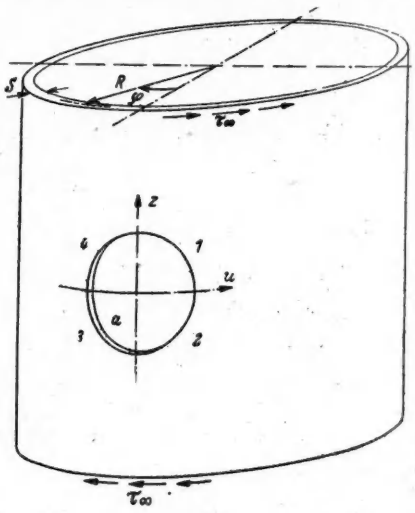


Fig. 1

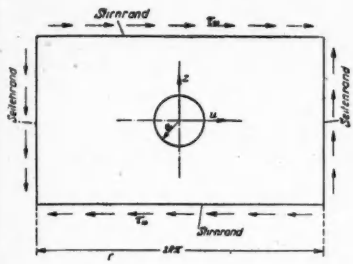


Fig. 2

$$\left.\begin{aligned}\sigma_r &= \tau_\infty \left(1-4\frac{a^2}{r^2}+3\frac{a^4}{r^4}\right)\sin 2\alpha \\ \sigma_t &= \tau_\infty \left(-1-3\frac{a^4}{r^4}\right)\sin 2\alpha \\ \tau &= \tau_\infty \left(1+2\frac{a^2}{r^2}-3\frac{a^4}{r^4}\right)\cos 2\alpha\end{aligned}\right\} \quad \ldots\ \ldots \quad (1)$$

The circumferential stress is also calculated:

$$\sigma_u = \frac{\sigma_r+\sigma_t}{2} - \frac{\sigma_r-\sigma_t}{2}\cos 2\alpha + \tau \sin 2\alpha \quad \ldots\ \ldots \quad (2)$$

And by substitution:

$$\sigma_u = -\tau_\infty \left[2\frac{a^2}{r^2}\sin 2\alpha\left(-2\frac{a^2}{r^2}+3\frac{a^4}{r^4}\right)\sin 4\alpha\right] \quad (3)$$

Consider now the plate to be rolled into a cylinder again. As bending stresses are produced in it, it is required to determine what additional forces must be brought to bear on the surface so as to maintain a true cylindrical form. As in this case no bending stresses exist, the condition of equilibrium for displacement normal to the wall is:

$$\sigma_u + \frac{p\,R}{s}, \quad \ldots\ \ldots\ \ldots \quad (4)$$

p being the internal pressure, and s the wall thickness. This formula is, in the absence of bending moments, of general validity (see D. Thoma in *A. Foeppl zum 70. Geburtstag*, Springer, Berlin, 1924, p. 42). Using equ. (4), it is clear that under the action of a supporting internal pressure,

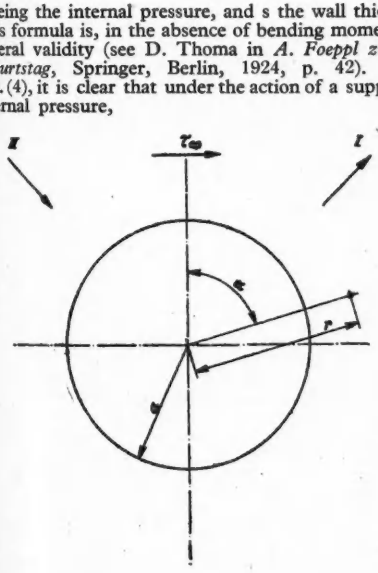


Fig. 3

$$p_{st} = -\tau_\infty \frac{s}{R}\left[2\frac{a^2}{r^2}\sin 2\alpha + \left(-2\frac{a^2}{r^2}+3\frac{a^4}{r^4}\right)\sin 4\alpha\right] \quad (5)$$

an exact cylindrical shape could be maintained. In reality, this internal pressure does not exist, i.e., actual loading differs from that which would prevent bending by

$$p = +\tau_\infty \frac{s}{R}\left[2\frac{a^2}{r^2}\sin 2\alpha + \left(-2\frac{a^2}{r^2}+3\frac{a^4}{r^4}\right)\sin 4\alpha\right] \quad (6)$$

To obtain the bending stresses which are present in a cylinder in torsion, an internal pressure according to equ. (6) must be added to the unstressed cylinder. This causes a deformation of quadrants 1 and 3 outward (Fig. 1), and quadrants 2 and 4 inward.

These conclusions were confirmed by tests on a hollow cylinder of rubber, and the deformation can be seen in Fig. 4. (A sheet of celluloid can be used instead of rubber if desired.—Ed.).

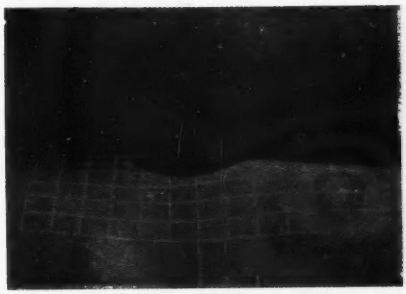


Fig. 4

PART II.

A rigorous solution of the bending stresses due to the pressure p involves considerable mathematical calculations. Therefore, a less exact estimation will be made by assuming the cylindrical surface in the proximity of the cut-out to be a flat plate. The following symbols are introduced in addition to those used in Part I.:

w = deflection normal to plate.

E = Young's modulus.

$N = \frac{Es^3}{12\,(1-\nu^2)}$ = flexural rigidity of plate.

$\nu = 0.3$ = Poisson's ratio, $\zeta = r/a$.

The theory of bending of plates is assumed to be known. By writing Δ for

$$\frac{\partial^2}{\partial \zeta^2} + \frac{1}{\zeta^2}\frac{\partial}{\partial \zeta} + \frac{1}{\zeta^2}\frac{\partial^2}{\partial \alpha^2}$$

the differential equation of our problem becomes:

$$\Delta\Delta\, w = +\frac{sa^4}{R}\frac{\tau_\infty}{N}\left[2\zeta^{-2}\sin 2\alpha + (3\zeta^{-4} - 2\zeta^{-2})\sin 4\alpha\right] \quad (7)$$

(The original paper consists of twenty equations of purely mathematical interest, which are, therefore, omitted from the present abstract. The author makes reference to his earlier work in *Z. angew. Math. Mech.*, 1926, p. 484, and to the books, *Fluegge, Statik* and *Dynamik der Schalen*, Springer, 1934, and *Nadai, Elastische Platten*, Springer, 1925. As to the physical side of the picture, a "support" of the plate in finite distance a from the edge of the cut-out is involved. It is here, that we pick up the thread again.—Ed.).

Approximately:

$$\beta \approx \frac{2\pi}{4\sqrt{3(1-\nu^2)}}\frac{\sqrt{s\,R}}{a} \approx 4.88\frac{\sqrt{s\,R}}{a} \quad \ldots\ (8)$$

The bending moment m_{to} in circumferential direction at the edge of the cut-out:

$$m_{to} = + \frac{S}{R} a^2 \tau_\infty [A^2 \sin 2\alpha + A4 \sin 4\alpha] \quad .. \quad (9)$$

The values A_2 and A_4 for various values of β are given in Table I.

TABLE I.

β	A_2	A_4	$\alpha\ m_{t\ max}$
2	+ 0.2204	— 0.0030	45° 47′
3	+ 0.3649	+ 0.0138	42° 52′
4	+ 0.4589	+ 0.0220	42° 18′
5	+ 0.5280	+ 0.0263	42° 13′
10	+ 0.1310	+ 0.0323	42° 31′

This shows that A_4 is only about 1/20 of A_2. In the last column the angle α is given for which the bending moment m_t reaches its maximum. To obtain the bending stresses σ_{tB} at the surface $\pm$ s/2, the bending moment m_t is to be divided by 1/6 s^2. Thus:

$$\sigma_{tBo} = \pm \frac{6\,a^2}{s\,R} \tau_\infty [A_2 \sin 2\alpha + A_4 \sin 4\alpha] \quad .. \quad (10)$$

By substituting in equ. (10) $\alpha\ m_{tv\ max}$ for α, we obtain the maximum bending moment. If we replace $\alpha\ m_{t\ max}$ by the approximate value $\alpha = 45°$, we obtain $\pm \frac{6\,a^2}{s\,R} \tau_\infty A_2$, whereby the error, as compared to the maximum value, is less than 0.5%.

In equ. (1) of Part I. the stress σ_t has its maximum value at $\alpha = 45°$. This circumferential stress over the wall-thickness is evenly distributed and equal to $\pm 4 \tau_\infty$. If we superimpose now the bending stress, we obtain at the surface $\pm$ s/2 the maximum stress

$$\sigma_{max} = \pm \tau_\infty \left[4 + A_2 \frac{6\,a^2}{s\,R} \right] \quad .. \quad .. \quad (11)$$

For example: a = 10 cm.
R = 62.5 cm.
S = 1 cm.

According to equ. (8) $\beta \cong 3.85$.

From Table I. we find the value of A_2 for $\beta = 4$ the nearest value, and we obtain $\sigma_{max} = 8.4\ \tau_\infty$.

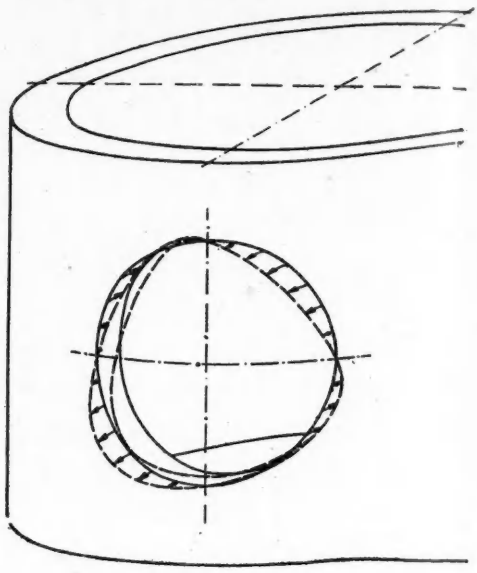


Fig. 5

This value appears to be extremely high; one could think that the value of β is not suitable. However, according to the testimony of Prof. H. Peters, of Muenchen, at the edges of cut-outs in monocoque fuselages, stress concentrations have been measured of 10 times the mean torsional shear stress of the full section. Figs. 65 and 66 of the Paper by Thum & Petri (VDI, *Forschungsheft* 409) show a stress concentration factor of 9. However, the different cross-section used by Thum & Petri does not allow direct comparison with our case.

Fig. 5 shows the estimated deformation of the edge of cut-out. The deformation shown is greatly exaggerated to illustrate the theory.

PROBLEMS OF NAVAL AIRCRAFT.

By PROF. DR. ING. H. EBNER, Hamburg. (From *Luftwissen*, Vol. 9, No. 5, May, 1942, pp. 142-154).
(*Continued from p.* 359).

Stability in Taxying.

In the preceding chapter the speed of the aircraft relative to water was assumed to be zero. When taxying, hydrodynamic forces come into play which may considerably alter the conditions for stability.

Fig. 9 illustrates this on a scale model-test of a two-float seaplane. The restoring moments, M, are plotted against the tilt angle φ at various speeds Ve. It should be remembered that when taxying with side wind, the direction of motion and the longitudinal axis of the aircraft enclose an angle the magnitude of which corresponds to an equilibrium of hydrodynamical and aerodynamical lateral forces. Such an angle of yaw (10° in Fig. 9) disturbs the flow along the float and thus reduces the hydrodynamic lift. Therefore, the limits of lateral stability in presence of yaw are rather reduced.

Another problem of stability arises when naval aircraft has to alight owing to engine trouble, and is drifting without control. In such case of emergency first it is necessary that the aircraft weather-cocks, i.e., swings head to wind. If it is now drifting backward at greater speeds it is liable to capsize over the stern. To avoid this drag anchors are being used, the ropes of which are attached to the aircraft at such a height as to create an additional erecting moment. Apart from achieving that the aircraft weather-cocks, directional stability is required for take-off too. This is possible by correct shaping of the stern, however, only in connection with the control characteristics on the water.

Problem of Stress.

The Alighting.

The magnitude of the alighting impact is dependent on the normal speed with which the float (or hull) bottom hits the water. The speed is composed of the flight speed, the wind, and the wave speed normal to the water surface.

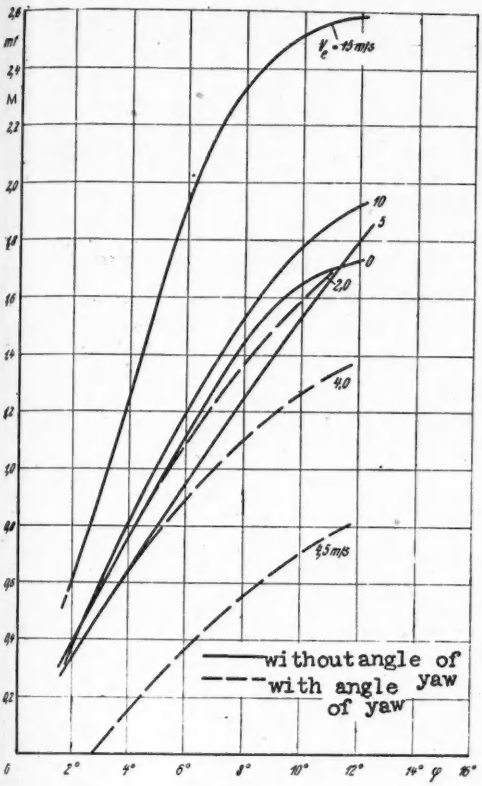


Fig. 9

To make the process of alighting accessible to calculation, many simplified assumptions must be made. In the first place it is assumed that the float which has no forward speed hits a plane water surface, whereby the length of immersed parts as well as the angle of V-bottom remain constant. Pabst has dealt with the theory of the alighting impact (DVL- Jahrbuch 1930), in which the aircraft is replaced by a system of masses connected by a spring. The strength of the spring must either be assumed or determined from experiments. Another simplification is involved in the definition of the "co-accelerated watermass." By accelerated motion of a body in water, kinetic energy is imparted to displaced water masses which, in turn, act as pressure on the body. The energy of the water can now be replaced by the energy of a constant mass of water, which is dependent on the shape of the body and has the same acceleration as the body. For a V-bottom float dipped into water the co-accelerated watermass constantly increases and is always equal to a semi-cylinder, the diameter of which is 2c, the wetted breadth. The latter can be found by Wagner's theory (ZAMM, Vol. 12, No. 4, 1932). The force acting on the float bottom can be derived from the moment of the co-accelerated watermass.

For purposes of calculation the aircraft is divided into two masses, with a spring between them (Fig. 10). The lower mass m_s represents the float, the upper mass

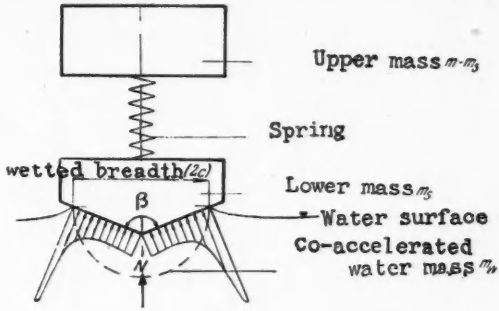


Fig. 10

$m-m_s$ the rest of the aircraft, and the massless spring stands for the elastic float struts. The co-accelerated water mass m_w is also indicated.

With this system the alighting impact N can be calculated for two extreme cases: for the elastic flat bottom, at which m_w is constant and for the rigid V-bottom, where the hardness of the spring is taken as infinite. The formulae are as follows:

Elastic flat bottom:

$$N = \sqrt{\frac{\mu^2(1-\eta)}{(1+\mu)(\eta+\mu)}}\ Vo\ \sqrt{m\,k}$$

Rigid V-bottom:

$$N = \frac{\sqrt{2\pi\mu}}{(1+\mu)^3}\ Vo\ \frac{\beta}{\pi-\beta}\ \sqrt{m\,l\,\rho_w}$$

Vo = Normal speed
l = Length of wetted float
k = Spring constant

$\mu = m_w/m$
$\eta = m_s/m$

N is a maximum for the V-bottom at $\mu = 1/5$, provided that the full breadth was not previously wetted.

The pressure distribution shown in the figure can be calculated from the change with time of the flow-potential according the previously mentioned theory of Wagner.

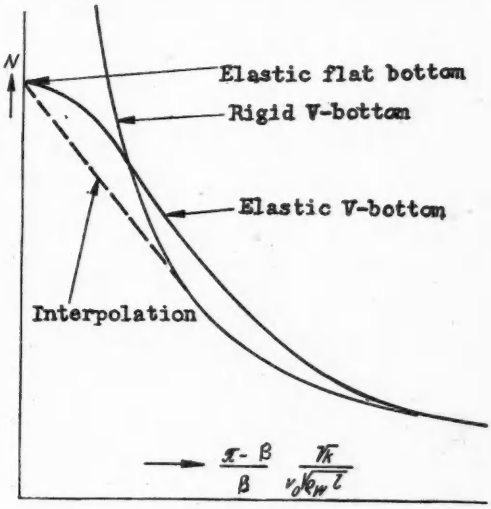


Fig. 11

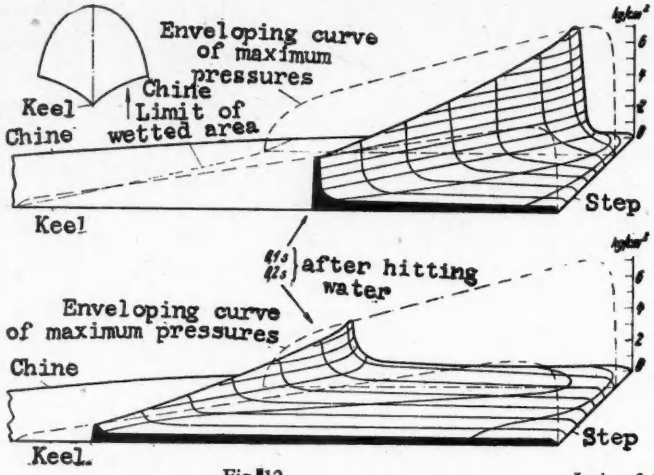


Fig. 12

The actual float lies somewhere between the above extreme cases, and the simultaneous expression of the two influences is more difficult. However, Sydow and Schmieden succeeded in presenting a solution (Jahrbuch 1938 der *Deutschen Luftfahrtforschung*) which is valid for all systems with equal Canchy numbers. Using this method a comparison has been made of the two extreme cases with an elastic V-bottom float and the result is plotted in Fig. 11. It is surprising that in the region of greater acuteness of the V, the forces N are greater on the elastic V-bottom than on the rigid one. The interpolation indicated in Fig. 11 by the dotted line is, therefore, not permissible. The theoretical results were confirmed by drop tests.

The Bottom Pressure.

In order to confirm the theoretical pressure distribution and to have a basis for the strength requirements, measurements have been made on starting and alighting naval aircraft by means of the DVL pressure recorder. This instrument is fastened by its flange to the float bottom, so that its membrane of 5 cm diameter can deflect under the action of pressure.

This, the membrane, measures the mean pressure over its area, and the measured pressure peaks, are, therefore, smaller than those theoretically found. Fig. 12 shows the result of measurements made by a number of suitably placed instruments. The two diagrams are 1/10 of a second apart from each other, and half of the bottom only is shown as the pressures are assumed to be symmetrical about the keel. As can be seen, maximum pressures of about 7 kg/cm² were encountered. Keeping in mind that average pressures were recorded, Wagner's theory gives satisfactory agreement with the test results.

Loading of the Float-Struts

Due to inaccuracies in the strength requirements and to simplify the stressing, it is desirable to know the actual loads and stresses in strut members of the float mounting. Fig. 13 is made at the beginning of alighting and shows two stress peaks sustained by the strut in question.

It is of interest to know how far actual stresses approach ultimate failing stresses, or to know the difference over their calculated values.

For problems of fatigue, however, or other design questions a statistical approach would be necessary as outlined by Gassner (*Luftwissen*, Vol. 1939, No. 2). The integral curves from many measurements are then compared with the Woehler-curve of the material and the fatigue strength of the member reduced.

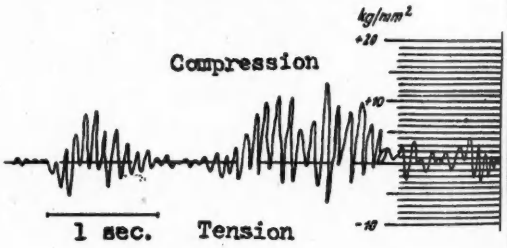


Fig. 13

RECTANGULAR PLATES UNDER UNIFORMLY DISTRIBUTED FLUID PRESSURE.

By M. NEUBERT and A. SOMMER. Report of the Dornier Werke, Friedrichshafen a.B. (From *Luftfahrtforschung*, Vol. 17, No. 7, July 20th, 1940, pp. 207-210).

THE knowledge of stresses due to fluid pressure is important to the designer of aircraft, as such pressure is often encountered by structural parts. These may be in the first place hulls or floats of seaplanes at take-off and alighting, but also fuel tanks, when subjected to high internal pressure due to acceleration, have to be considered.

Two calculations—by A. and L. Foeppl (Drang und Zwang) and by Hencky (Z. angew. Math. Mech., Vol. 1, 1921)—are available and will in the present report be compared with test results in order to establish their applicability and reliability.

Foeppl's Method.

The solution presented by Foeppl is an approximation, and gives simple formulae for practical use, which are valid only if the deflection of the plate is considerably greater than the sheet thickness. For gauges used in aircraft this will always be the case.

Fig. 1 shows the plate and notation (a and b in cm). Further in the formulae below:

p = uniformly distributed fluid pressure in kg/cm²
E = Young's Modulus in kg/cm²
s = sheet thickness in cm

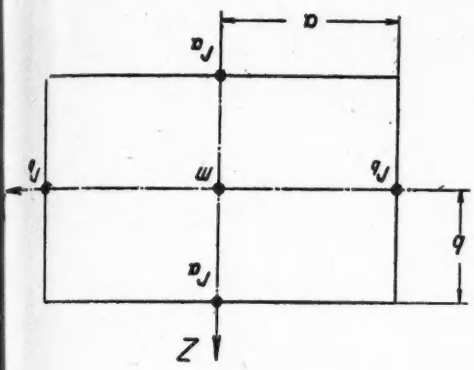


Fig. 1

Deflection at point m: $f = n_1 \sqrt[3]{\frac{p\,a}{E\,s}}$

Stresses at point m: $\sigma z_m = n_2 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

$\sigma y_m = n_3 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

Stresses at point r_b: $\sigma z_{r_b} = n_4 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

$\sigma y_{r_b} = n_5 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

Stresses at point r_a: $\sigma z_{r_a} = n_6 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

$\sigma y_{r_a} = n_7 \sqrt[3]{p^2 E \frac{a^2}{s^2}}$

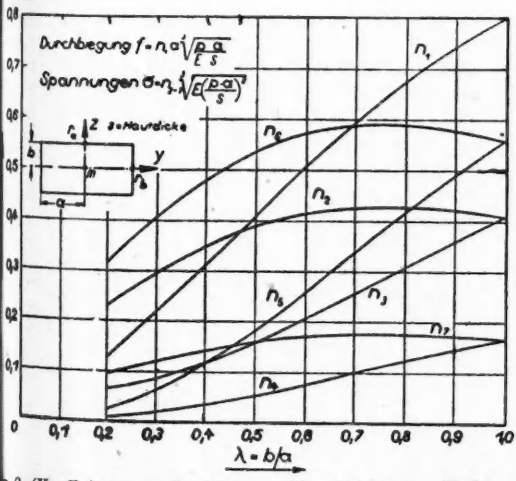


Fig. 2 (Koeffizienten = Coefficients; Durchbiegung = Deflection; Spannungen = Stresses; Hautdicke = Sheet thickness.

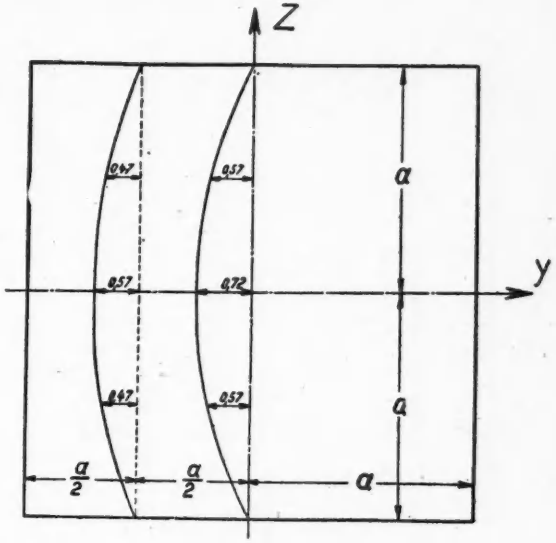


Fig. 3

The coefficients n_1 to n_7 are plotted in Fig. 2 against the ratio $\lambda = \frac{b}{a}$. The greatest stresses, therefore, occur at point r_a in z direction. In the case of the quadratic plate the greatest stress is not at m as stated in Drang und Zwang, but at the middle of each line bounding the plate. Foeppl has accepted this correction and intends to incorporate it in a new edition of his book.

Numerical Example.

Plate size 60 × 60 cm, Sheet thickness s = 1.4 mm, E = 740,000 kg/cm², Loading p = 20 t/m².

Estimation of deflection f at the middle of the plate:

$$f = 0.8a \sqrt[3]{\frac{pa}{Es}} = 0.8 \times 30 \times \sqrt[3]{\frac{2 \times 30}{740{,}000 \times 0.14}} = 1.99 \text{ cm.}$$

Estimation of maximum stress

$$\sigma z_{r_a} = \sigma y_{r_b} = 0.56 \sqrt[3]{E\left(\frac{pa}{s}\right)^2} =$$

$$= 0.56 \sqrt[3]{740{,}000\left(\frac{2 \times 30}{0.14}\right)^2} = 2700 \text{ kg/cm}^2$$

Hencky's Method.

Solutions were arrived at by means of the differential calculus. Results for the quadratic plates only are presented as the numerical work for solving other cases is considerable, and has not been done as yet.

The deflection of the quadratic plate:

$f = n_1 a \sqrt[3]{\frac{pa}{Es}}$. The coefficients n_1 re given in Fig. 3.

Stresses:

$$\sigma_z = n_2 \sqrt[3]{E\left(\frac{pa}{s}\right)^2} \qquad \sigma_y = n^3 \sqrt[3]{E\left(\frac{pa}{s}\right)^2}$$

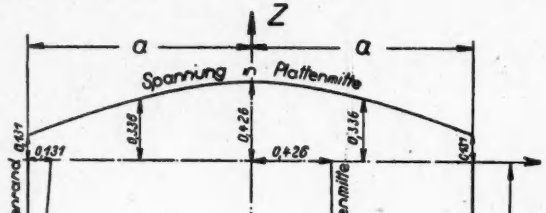


Fig. 4 (Spannung in Plattenmitte = Stress at middle of plate; Spannung am Plattenrand = Stressat bounding line of plate).

The coefficients n_2 and n_3 are shown in Figs. 4 & 5.

For the above example the estimation of deflection and maximum stress is as follows:

$$f = 1.99 \cdot \frac{0.72}{0.80} = 1.8 \text{ cm}$$

$$\sigma_{z_{r_a}} = \sigma_{y_{r_b}} = 2700 \cdot \frac{0.436}{0.56} 2100 \text{ kg/cm}^2.$$

Comparison of Test Results with Calculation.

Plates of 60 × 60 cm, and of various sheet thicknesses have been tested. The test set-up is shown in Fig. 6. The sheet is riveted to two frames, one on either side, and screwed to the pressure vessel. On top of the double frame there was a strong lid, strongly reinforced by means of numerous ribs. So was the vessel itself to exclude the possibility of its deformation.

Measurements have shown that the estimated plate deflections and stresses were in agreement with the tests within the elastic range of the material, fully justifying the assumptions on which the theory was based. According to present German design requirements this is also the practical range. Plastic deformation, however, starts at relatively low values, and then the calculation gives wrong results, too small deflections, and too high stresses. But as sheet in tension is capable of sustaining large plastic deformations, the load carrying capacity of the plate exceeds by far the calculated value. The permament deformation, i.e., the deformation which remains after unloading, is also considerable.

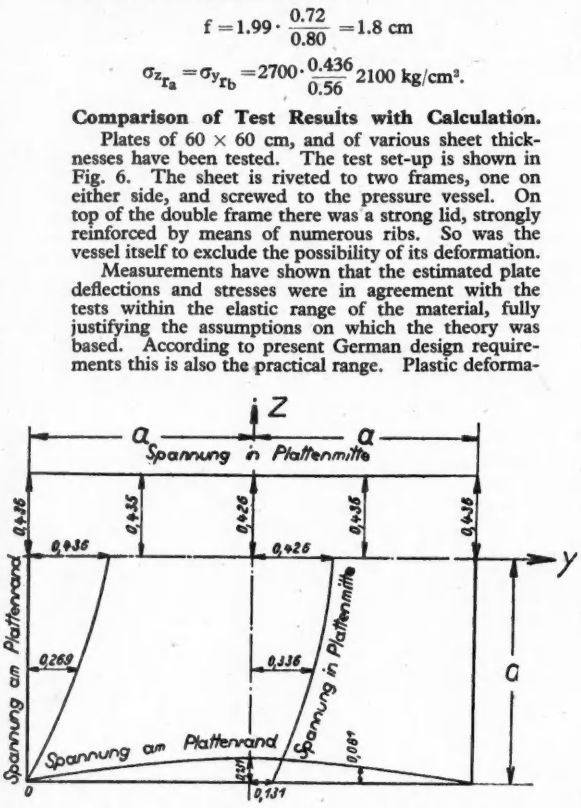


Fig. 5. (Spannung in Plattenmitte = Stress at middle of plate; Spannung am Plattenrand = Stress at bounding line of plate).

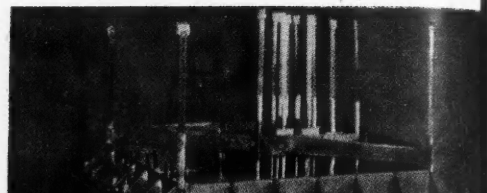


Fig. 6

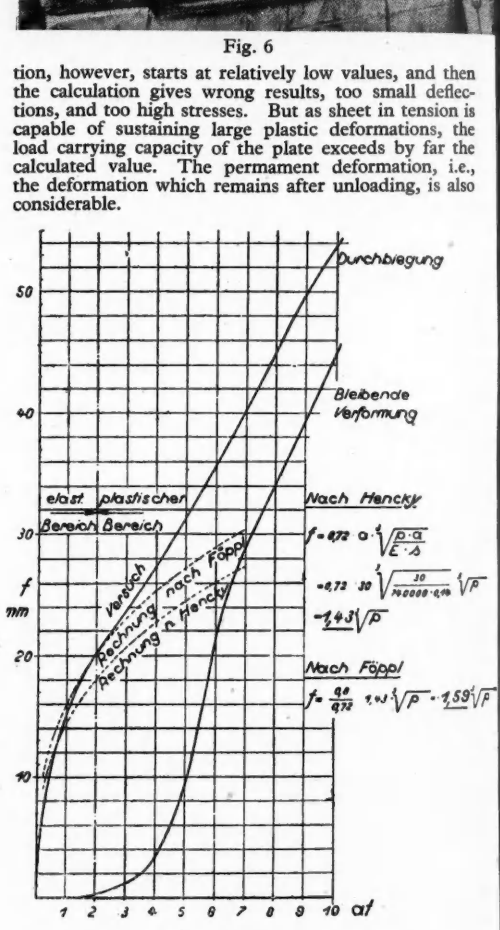


Fig. 7

(Elast. Bereich = Elastic range; plastischer Bereich = Plastic range; Versuch = Test; Rechnung nach = Calculation; Durchbiegung = Deflection; Bleibende Verformung = Permanent Deformation).

The test results for a plate of 60 × 60 cm and a sheet thickness of 1.4 mm. are compared with the estimated values in Fig. 7. All values refer to the plate middle. The plate failed at 11.2 ats., and the elastic range up to 2 ats. appears to be rather limited when compared with the loading at failure. At a stress of $\sigma_2 = 4{,}400$ kg/cm² the failing load is, according to Foeppl,

$$P = \sqrt{\frac{\sigma^3 s^2}{n^3 E a^2}} = \sqrt{\frac{4400^3 \times 0.14^2}{0.56^3 \times 740{,}000 \times 30^2}} = 3.8 \text{ ats.},$$

according to Hencky,

$$p = \sqrt{\frac{4400^3 \times 0.14^2}{0.44^3 \times 740{,}000 \times 30^2}} = 7.14 \text{ ats.}$$

The actual failing load is much higher for the reasons given above. In the elastic range Foeppl's solution gives lower values as compared with Hencky, and is on the safe side.

Summing up, it can be stated that both methods give correct results within the elastic range, and the load carrying capacity of the plate is about 50 to 100 per cent. higher than can be calculated by using both methods for the elastic range.

THE ELECTRIC DRYING OF GRASS.

By Dr. R. BERNSTEIN, Baden. (From *Brown Boveri Review*, Vol. XXVIII, No. 4/5, April/May, 1941, p. 75-83.)

SWITZERLAND has drawn up a special cultivation programme under which it is hoped the country will become as self-supporting as possible. A most difficult problem is to get sufficient fodder rich in albumen to feed the cattle, which is essential to keep up the meat supply and dairy products. All the countries which are at war or affected by the war, in the part of Europe where rains are plentiful, are faced by the same problem. All these countries hope to remedy the lack of fodder for the cattle by drying green fodder grown at home. Thanks to the abundance of water power during summertime, this source of heat can be used in Switzerland for the production of high-grade fodder most advantageously.

Grass contains, when fresh, only a little under 20% of dry substance containing nourishment, and sometimes no more than 10% thereof. The content of 80% water must be eliminated in order to conserve the nourishing substances for a considerable time. The remaining amount of water in the dry stuff should not exceed 10% on an average, if mould is to be avoided. From 100 kg. of grass, 80 kg. or more of water must be drawn off. As by pressing a good deal of the valuable juices would become lost, the only thing to do is to eliminate water in the form of steam. The heat of evaporation for a kg. of water is about 600 calories under vacuum, and rises to over 700 calories when atmospheric air is freely admitted. In practice, up till now no less than 1,000 calories have been attained in the drying apparatus used, generally this figure has been exceeded. With such consumption about 75 kWh. are required for producing 20 kg. dry grass out of 100 kg. fresh grass.

In winter, when there is no fresh fodder, the digestible parts contained in the dry grass as compared to the indigestible ones of hay have diminished. Albumen is the substance chiefly lacking. In normal times the low grade fodder was supplemented by some high grade fodder containing more nourishment, termed concentrated fodder. To-day, however, not only the import of such fodder as oats, barley, Indian corn, oil-yielding fruits, etc., is very difficult, but that of grain for making bread as well, so that feeding the cattle with granular fodder cannot be thought of. Therefore, it is of greatest importance to avoid as far as possible any loss in the nutritious value of fodder during harvesting and artificial drying. Since long it has been found that a concentrated form of fodder can be obtained by the artificial drying of green grass. This fodder contains not only as much nourishment as ordinary concentrated fodder, especially albumen, but is characterised by fermentations, which help digestion, especially Vitamin A and Carotin, in a proportion which is not much less than what is found in fresh grass. Therefore, "dry grass," as the product of artificial drying is known, may be considered equal in value to other concentrated fodders. The value of the fresh grass which has to be cut to obtain 100 kg. of dry grass can be evaluated at Fr. 9 on the average, while the peacetime price of concentrated fodder products is about Fr. 15 per 100 kg., and at its present price of Fr. 30 is hardly obtainable. Great efforts to produce as much dry grass as possible are fully justified and expenditure to this end will also be justified when we return to normal conditions of life. However, the drying process has to be rationally carried out, if it is to be something more than an emergency measure to be applied only in present periods of abnormally high cost of living.

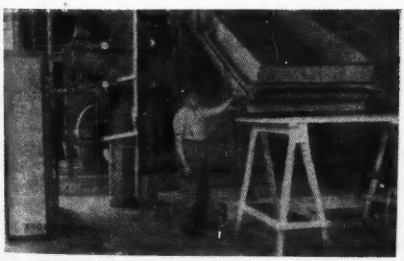


Fig. 1. Experimental grass dryer with electric heating and recuperation of heat. Seen from the side.

The usual methods of drying grass.

The former system of drying apparatus was not at all satisfactory, although there were several more or less practical apparatuses available. The many efforts to

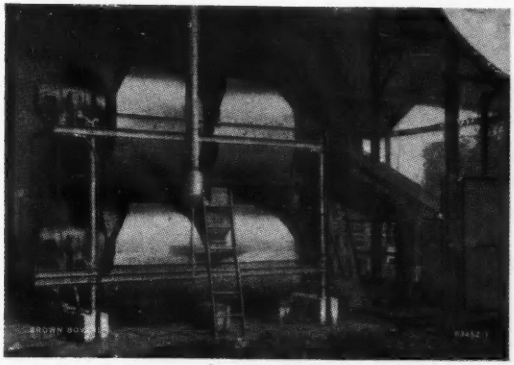


Fig. 2. First electric grass dryer ever built with recuperation of heat. An experimental apparatus for 50-80 kg of dry grass per hour. Up to 50% of the energy utilized to obtain the same results in other processes is saved here by heat recuperation.

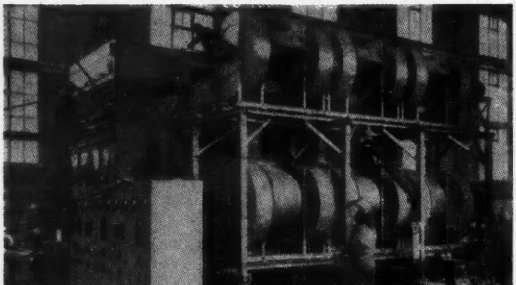


Fig. 3. Experimental grass dryer with electric heating and recuperation of heat, seen from delivery side.

reduce the three cost-factors: capital, attendance and power consumption had no satisfactory results until the new drying apparatus using heat-recuperation was designed. The primitive hurdle dryers were inexpensive to build, as they were simple boxes with sieve bottoms on to which the grass was shaken in a layer of varying thickness and subjected to a flow of hot air passed through the dryer. But these apparatuses were expensive on the head of attendance and on that of power consumption. Hurdle dryers in several levels result in better utilization of the power input, but require still more outlay for attendance. Neither special mechanical conveying devices nor attempts to bring down the heat consumption achieved a lowering of the consumption of 1,000 calories per kg. of water evaporation.

The amount of dry grass to be obtained from 100 kg of green grass fluctuates between 12 and 35 kg., 18 kg. being an average value. The fixed costs of the drying process may be calculated on the assumption that an electrically operated grass dryer would cost, at present, Fr. 100,000, including the housing and the accessories. But even if a cheap tariff of 2 centimes per kWh is assumed, the power costs outweigh by far all the other cost factors. This brings heat recuperation applied in new grass dryer to an outstanding effect.

It should be mentioned that there exists still another method of preserving fodder, which is cheaper than artificial drying, and which is very recommendable—the ensiling. It consists in enclosing the chopped grass in containers from which air is excluded. The fodder is subjected there to a fermentation process, by which it is preserved. The contents of the containers, which have to be provided, is very considerable; so that the costs of the two processes are about equal. However, dry grass fetches a somewhat higher price on the market than does silos fodder, because the latter is not suitable for cows, the milk from which is used to make cheese.

Grass dryers with heat recuperation.

As the fresh grass is very apt to clog openings and susceptable to temperature, it is impossible to use containers under vacuum or under pressure. Thus, air heated electrically to an appropriate temperature is driven under very low pressure through the drying chamber of the new drying apparatus, built by Brown Boveri in collaboration with U. Amman, machine factory of Langenthal, Switzerland. The vapours and the hot air after having passed the first chamber of the dryer, passes an apparatus working as a condenser through which fresh air is driven by a blower. The steam and the other vapours are condensed, and the fresh air absorbs the heat set free by condensation. The heated fresh and dry air is driven through a second drying chamber, into which the grass has been brought after having passed the drying process in the first chamber. A considerable quantity of heat is utilized in this way in the second drying chamber.

The first test of drying according to such a recuperation method reduced the necessary quantity of heat to a much higher degree than expected. The grass to be dried is carried successively through the two chambers on several superimposed endless sieve belts. The latter are so designed that no adherence of material at the driving organs occurs. The thermal exchange process is achieved by an appropriate drive of the hot air over the grass transported by the sieve belts.

The first apparatus built according to the fundamental principles explained in the preceding paragraphs, has been tested in 1940 during several months on the property of the Zentralschweizerische Kraftwerke, in Emmen. It is shown in Figs. 1-3. The result of one test is given in Table I. The product had excellent quality, as the grass was not spoiled by high drying temperatures, which damage the fodder produced.

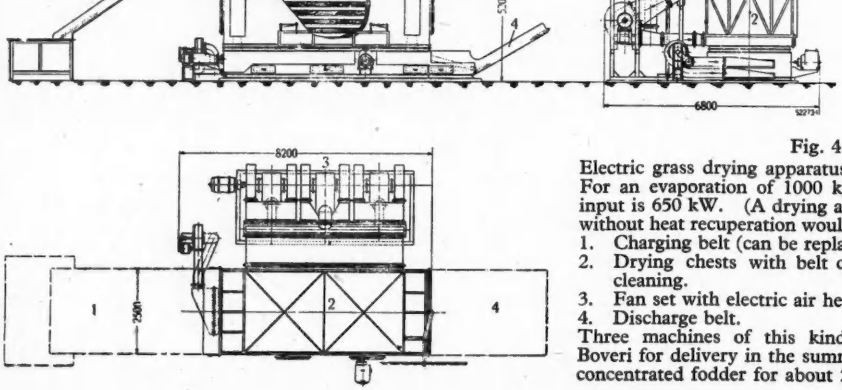


Fig. 4

Electric grass drying apparatus with heat recuperation. For an evaporation of 1000 kg/h of steam the power input is 650 kW. (A drying apparatus of same output without heat recuperation would take at least 1200 kW).

1. Charging belt (can be replaced by grab hoist).
2. Drying chests with belt conveyor and doors for cleaning.
3. Fan set with electric air heater and heat exchanger.
4. Discharge belt.

Three machines of this kind ordered from Brown Boveri for delivery in the summer of 1941 will provide concentrated fodder for about 2000 head of cattle.

TABLE I.

(Test carried out on August 26th, 1940.)

1. Quantity of grass	3.33 kg/h.
2. Water content	81.5%.
3. Quantity of dried grass ..	67 kg/h.
4. Evaporated water	266 kg/h.
5. Ratio of dried to fresh grass ..	1 : 5.
6. Final water content	about 10% of the dry substances.
7. Energy consumption including motors	200 kW/h.
8. Specific energy consumption	650 kcal/kg H_2O.

Energy input for heating and motor drive was about 700 calories per kg. water, that is 30% less than with dryers of previous design. Table II. gives the results of investigations into the albumen content of the products of some other tests. Even better figures than given in Table I. may be attained if all the different elements in the machine are properly designed. The saving of costs of electric energy would allow of compensating for quite a certain increase in the cost of the plant if this were necessary. However, the latest experience gained shows that the plant is not made more expensive by the recuperation equipment. The saving made in power input with the apparatus with recuperation is pure gain. The effect of this saving is further increased because it is possible to dry with such an apparatus other products besides grass, for example, fruit or vegetables or freshly cut cereal, and thus to increase the yearly number of hours during which the machine can be used. Fig. 4 shows the latest design of the dryer. The feed belt is seen and the drying chamber subdivided into a number of accessible doors for cleaning purposes, the ventilation part with air heater and heat recuperator and the delivery belt are also visible. The latter is located above so as to facilitate delivery of the dry grass to a bale press. The pressing of the dried grass into bales has proved a practical measure to prevent the hygroscopic material absorbing humidity from the surrounding air.

TABLE II.

Date of Test.	Albumen content. % Fresh.	Albumen content. % Dry.	H_2O content of test. %	Max. temp. °C	H_2O content at beginning. %	*Notes.*
23. 9.40	22.95	22.54	—	145	85.6	Fresh dew-wet grass.
11.10.40	—	23.16	10.6	145	75	Oats, fresh, wet by rain.
17.10.40	—	15.8	10.8	150	84	Ordinary fresh grass.
23.10.40	23	22.6	7.87	150	83.72	Rich fresh grass.
25.10.40	17.8	16.05	7	140/130	75.5	Clover mixed with grass, passed twice through the machine.
14.11.40	24	23.19	8.5	190/125	83.5	Fresh dew-wet grass, passed twice through the machine, high temperature at first pass.

The % content of albumen is referred to the dry substance.

PISTON RING PRESSURE DISTRIBUTION

By DR.-ING. M. KUHN. (From *A.T.Z. Automobiltechnische Zeitschrift*, Vol. 45, No. 3, Feb. 10th, 1942, pp. 62-67.)

THE discovery and introduction of the internal combustion engine has resulted in a very rapid development in machines utilizing the action of a piston. Design has been limited by the internal components of the engine, which has been subjected to ever increasing thermal and mechanical stresses. Of these internal engine components, the piston and piston rings are of particular importance and the momentary position of engine development is not seldom dependent upon the development of both of the components.

The piston ring is a well-known component and has been used in its present shape in the steam engine of the last century. Corresponding to its importance, the piston ring has been a rich field for creative activity and it is noteworthy that in spite of this the ring has maintained its shape through the many years. From the many and complicated designs which have been suggested as a packing between piston and cylinder wall hardly one suggestion has remained which does not resemble the original design of cast iron rectangular ring.

Material Development.

From the metallurgical aspect it is known that the graphite in the ring structure should be present in fine filiform distribution, for only in this form of graphite distribution maximum strength properties can be obtained. The occurrence of eutectic graphite in the structure is fundamentally undesirable because it is invariably associated with the formation of ferrite pockets, which, from the point of view of wear, are bad. The remaining carbon is bound as pearlite in iron and its appearance is identified as fine lamella sorbite. The quantity of graphite and structure formation, besides being dependent upon the smelt charge and the composition of slag is dependent also on the gas temperature, the mould material and the cooling speed of the cast. The speed of solubility of the iron in the surface is dependent upon the alloying constituents, as well as from the speed of heating. Manganese increases, while silicon decreases, the speed of solubility.

This reaction of the alloying constituents necessitates the uttermost care in their uses because silicon expedites the precipitation of graphite in the structure while it hinders the pearlite formation. A large addition of silicon necessitates, therefore, an addition of manganese to prevent excessive graphite formation.

Better carbide formation results in improved thermal and strength properties while better graphite formation results in improved running properties. The carbide forming elements are chromium, molybdenum, vanadium, tungsten, arsenic, antimony, while the graphite forming elements are silicon, nickel, titanium, copper and aluminium.

As has already been pointed out, the alloying of an element of the first group usually necessitates the

alloying of an element from the second group, if neither carbide formation nor graphite precipitation is to predominate in the structure and cause the ring to have either good running properties and bad thermal and strength properties or the opposite effects.

In addition, it is necessary that phosphorous be alloyed to the ring. Although a phosphorous addition affects unfavourably the strength properties of the ring, the running properties are improved by its addition owing to the possibility of the formation of uniform phosphide pockets, thereby improving the adherence properties of the oil film.

Manufacturing Processes.

In general, the modern method of manufacturing piston rings is the individually cast method, whereby the blanks are moulded singularly and "heaped" cast. Cylindrical pot casting was popular for a long time, but its use has declined in spite of the fact that a very fine uniform structure was obtained. The elastic properties of the cylindrical pot cast rings were not sufficient to withstand the conditions in the modern engine. The individually cast ring showed in this respect promise, in that it was possible to stress the ring to a high degree. In Germany, the centrifugal method of casting rings is seldom employed, but in the English-speaking countries the method is adopted.

Ring Stressing.

Piston rings can be pre-stressed by three methods. The first method is by hammering, a method which is seldom used in Germany. By this method, the ring is pre-stressed by stretching the inner fibres of the ring. The process is applied by rolling or by notch-impact. The second method of pre-stressing piston rings is the heat formed method, which is being used extensively today. In this method a ring which is entirely free of stresses is split and the gap thereby resulting is forced apart by a wedge. In this condition the ring is then annealed whereby the stresses are nullified. An external stress is then applied to the ring, and, in the closed condition, is machined to finished size.

The third method of pre-stressing piston rings is the out-of-round method. In this method ring blanks are used which are not circular but formed according to an involute. The ring is then cut by cutting away an appreciable piece of the ring to form the gap. The ring is then pressed together and finish turned to a circle. After machining, the ring will maintain its natural casting stresses. A heat treatment follows which prevents any subsequent loss in stress due to the temperatures occurring under engine operation conditions.

Heat formed rings result in a uniform ring pressure distribution throughout the ring circumference, *i.e.*, a round radial pressure pattern. Out-of-round rings can be given any ring pressure pattern and it is possible to make the rings with high radial point pressures especially at the gap. It will be shown later that this is also possible with heat formed rings.

The Stress Distribution or Ring Pressure Pattern.

The question now arises as to what type of ring pressure pattern is desirable. Originally, one had endeavoured to obtain a uniform ring pressure pattern; this type of pressure pattern is very favourable, as, with it, the ring is pressed at all points with uniform pressure against the cylinder. One found from experience, however, that where the ring was subjected to maximum load, *i.e.*, at the gap, ring collapse took place.

It is desirable not to have a uniform pressure pattern when the ring is new, but after running the ring for some time under operating conditions a uniform ring pressure pattern should be attained. This means that when the ring is new the gap should have higher radial or point pressure than the rest of the circumference. In addition to this, with the continuous increase in engine speed, towards the higher speed range, certain processes occur, which cause sudden increases in piston ring blow-by and oil consumption. This process is termed piston ring flutter, a phenomenon, which probably occurs when the ring is excited by impulses in the range of its natural frequency. Reliable data is, however, not yet available to analyse this condition.

It can be shown experimentally, and the object of this article is to show, that when the ring pressure at the ring gap is high, the unfavourable condition of piston ring flutter and its undesirable effects can be moved outside the range of actual engine operational speeds.

The mean specific ring pressure of the ring is considered as a most useful criterion. The mean specific bearing pressure is calculated from the tangential force acting at the gap which is necessary to close the ring to the normal gap clearance. The measurement of the tangential force is made by means of a "Toledo" balance.

The following relation is obtained:

$$p_m = \frac{2 \cdot P_t}{h \cdot D} \quad \text{.....kg/cm}^2.$$

In which

p_m = mean specific ring pressure..kg/cm².
p_t = tangential force .....kg.
h = ring width .....cm.
D = ring diameter.....cm.

The results of stress obtained by means of the toledo method will agree with the actual mean surface pressure only when the ring pressure is evenly distributed over the whole ring circumference, which means that the radial pressure distribution pattern must be round.

The toledo stress and the stress obtained from the calculated mean specific pressure are not in agreement when the radial ring pressure is not evenly distributed around the ring circumference. The ring with uneven pressure patterns usually has high pressure points at the ring gap and at other parts of the ring circumference. The remaining sections of the ring may have low or even very low surface pressures. The whole ring cannot be considered stronger than that of its weakest section, and this section cannot be determined from the radial pressure distribution. It follows therefore, that for out-of-round rings, which have been stressed, the radial pressure distribution is important.

Measurement of Pressure Distribution.

To obtain actual measurements of radial pressures is not a simple matter and a method free of error does not exist. Many references are to be found in the literature relating to radial pressure distribution and the results obtained are often represented in a comprehensive manner. It is, however, noteworthy that the manufacturers usually recede for one reason or another, directly measurements are asked for. Most of the measuring methods give such unreliable data that they do not lend themselves to criticism.

It may be worth mentioning that the radial pressure distribution can be measured by the steel band method, by means of which the ring is stressed to its normal size by a thin steel band as shown in Fig. I. With the aid of calipers, the diameter of the ring is measured at various positions of its diameter. The out-of-roundness obtained is ring ovality which can be as much as one per cent. greater than the normal ring diameter. The

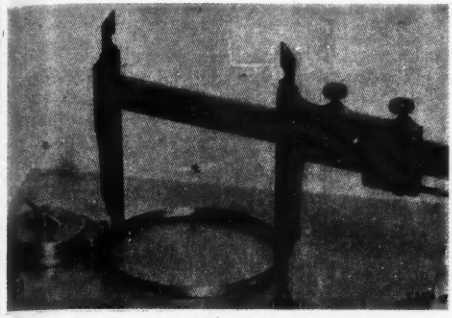


Fig. 1

positions of low ring pressure cannot be determined by this method of measuring ring pressures.

In spite of the fact that rings tested by this method can give pressure distributions which appear to be entirely satisfactory, they have failed when used in the engine. The failures were more frequent, the higher the loading of the combustion chamber. After very short operations in the engine, the faces of the rings at certain positions became discoloured and there is an increase in blow-by and oil consumption. Ring examination for high duty engines should be carried out, therefore, by some other method.

The obvious solution of the problem lies apparently in the use of a ring gauge, the inside surface of which has provision for a series of pressure gauge sockets or nipples. It is fully realized that the difficulty of the solution may be the finding and mounting of such pressure gauge sockets, which in the radial direction, permit the most accurate force indication with the smallest possible measurements, of the ring pressure. A relatively simple solution of the problem is described in the following arrangement and shown in Fig. 2.

The inner surface of a ring gauge is arranged with a number of steel rollers. Each roller can be loaded externally through an orifice. A pattern of the radial ring pressure distribution is obtained by the resistance of the roller to turning by manual operation. The latter is very troublesome and, in addition, roller turning by hand only gives subjective measurements.

A measuring method developed by the Mahle Company of Germany is shown in Fig. 3. This method permits objective measurement to be made by means of pressure gauges. The ring to be tested is inserted in a ring gauge with a 0.02 mm. greater diameter than the nominal ring diameter. The gauge points of sixteen clock gauges are in contact with points around the circumference of the ring. These pass through holes drilled radially through the ring gauge. By suitable screw adjustment the gauge points are pressed against the ring, so that all clock gauges show identical readings and, therefore, similar stress. The gauge points are then screwed further in against

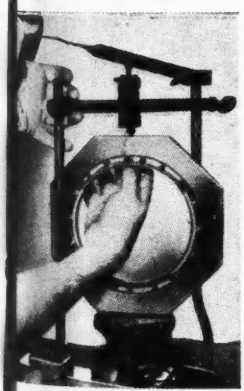


Fig. 2

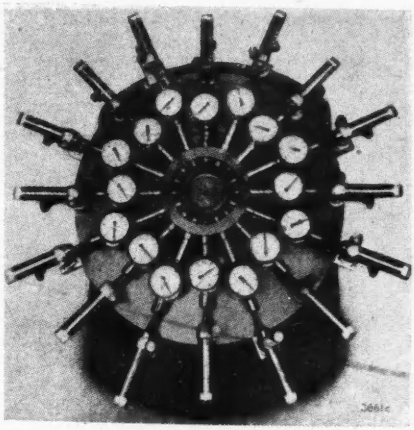


Fig. 3

the ring, by means of a "minimeter" to an amount equal to 0.01 mm. The readings obtained from the clock gauges present a pattern of the radial pressure distribution.

It has been found that this complicated apparatus cannot be used for practical purposes. The time required to take measurements is too lengthy, for when setting the clock gauges with equal pressure an alteration in the setting of one clock necessitates variation and adjustment of the other clocks. In addition, it is so complicated that it is impossible to design an apparatus suitable to test rings of different diameter.

A simpler apparatus for measuring pressure ring distribution is that which has been introduced and developed by Messrs. Alfred Teves in conjunction with the Zeiss Co. This apparatus operates on the piezo-electric principle and is shown in Fig. 4. An inducement to use the piezo-electric effect of a "Quartz" is that the quartz shows only a small tendency to deform in the direction of the applied load or pressure. The movement of the measuring point is practically zero. But it is a disadvantage that the piezo-electric effect occurs not at constant pressure but only during pressure changes.

The piston ring is inserted in a ring gauge of the apparatus. As the actual measuring procedure has been described by C. Englisch, the author does not consider it necessary to describe the apparatus at this stage. As an essential component of the equipment is a loop or cathode ray tube, it follows that a well equipped laboratory is desirable.

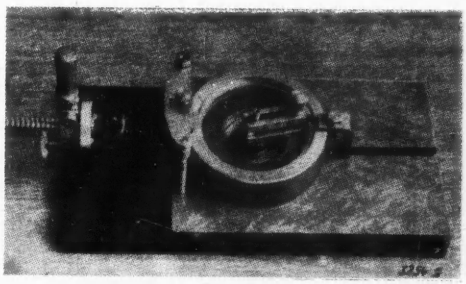


Fig. 4

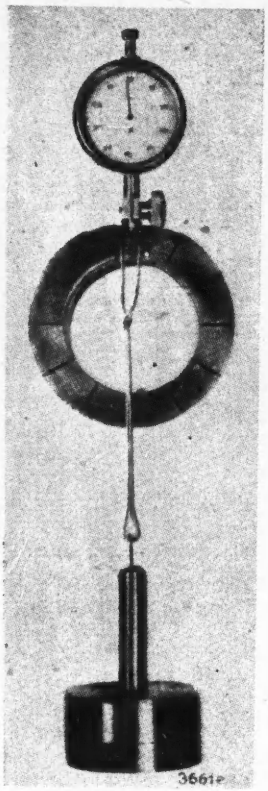


Fig. 5

Both the last two mentioned highly developed methods of testing piston rings are so elaborate and lengthy in time that they can only be used by specialist firms. The engineer who uses piston rings and requires further knowledge concerning their characteristics is, therefore, forced to use simpler methods of measurements such as the method which is about to be described.

The apparatus again consists mainly of a ring gauge, the nominal diameter of which corresponds to that of the piston ring to be tested. The ring gauge has a shoulder against which the piston ring is pressed. The gauge points of a clock gauge, which is graduated in 1/100 or 1/1000 divisions, pass through a hole in the ring gauge and rest upon the circumference or bearing surface of the ring. The apparatus is shown in Fig. 5. The front face of the ring has a series of notches which are situated on the right and left of the holes drilled for the gauge points. These notches are used to locate a hook, upon which weights can be suspended in order to load the ring bearing surface.

The procedure of measuring piston ring pressure distribution is as follows: After inserting the piston ring in the ring gauge, the ring gauge is adjusted to give a zero reading. A simple turning of the ring gives an indication of ring roundness, in the same way as is obtained with the light method. Everywhere, where the ring diameter does not correspond with the nominal diameter of the ring gauge, the clock indicates the occurring deviation. A satisfactory ring will be within the limits of from 0.002 to 0.005 mm. of its nominal dimension. It has been found that many rings measured deviate from the nominal diameter by as much as 0.03 mm. The weakest section of the ring is in the neighbourhood of the ring gap, while the ring gap itself retains the dimension of the nominal diameter.

Measurement of radial pressure distribution is carried out in the simple manner of hanging weights on the suspension scale pan attached to the hook. The weight is increased until the clock shows a deflection of 0.01 mm. The reduction in clearance at the gap is given by the product of π times the deflection and is of the order of 0.03 mm. This clearance reduction is unimportant, because even with rings of small dimensions the gap clearance is 0.15 mm., or greater. In this way the ring is measured at all sections and a pattern of the pressure distribution is obtained of surprising impressiveness. The suspended weight shows, in addition, the bearing or surface pressure occurring at each point with some approximation. By loading the ring until a reduction in diameter of 0.01 mm. occurs each point of the ring bearing surface moves a small distance during the measurement in the ring gauge. The frictional force, thereby set up, varies according to the position of the gap.

Should the point at which the measurement be made be near the gap, then with a reduction in diameter of 0.01 mm. the ring gap will not be reduced but the loaded ring ends will yield towards the centre of the ring. Hence, no friction between the ring and the gauge exists. Should the point at which the measurement is made be on the side opposite the gap, then a reduction in ring diameter can only occur with a corresponding reduction in ring gap. The friction which is set up between the ring and the gauge will be measured. It is for this reason, that the sum of all the pressures measured on the gap half of the ring are always smaller than those occurring on the opposite or other side of the ring. On the side opposite to the gap the measured pressures are higher than the actual ones by an amount equal to the frictional forces which are of unknown magnitude.

The occurring error in measurement or experimental error is not small, but it can be easily estimated from the position of the centre point to the pressure distribution curve, as the sum of the vectors of all the pressures must be zero.

Considered as a whole, the accuracy of measurement is probably greater than that obtained with the other methods of measuring ring pressure, described in this article, because the latter only gives a pattern of the ring pressure distribution and not an actual value of the measurement.

With a certain amount of experience, measurements with the apparatus can be made quite quickly. The time required to measure the ring takes five minutes or less. The radial pressure curves are so characteristic for a ring, that from the curves it can not only be ascertained whether the type of stress in the ring was obtained by heat-forming or the out-of-round method, but also indicate the source from which the rings were obtained, as the tools used in the manufacture give the ring product certain characteristics.

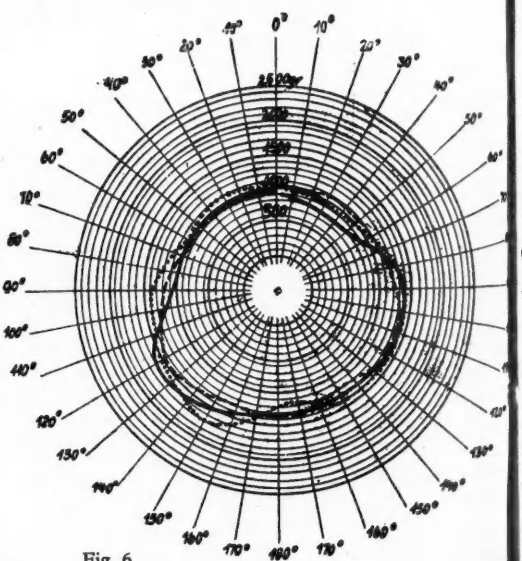


Fig. 6

Applications.

Fig. 6 shows the radial pressure distribution or pressure pattern of a heat formed ring, which had been used for some 8000 km. in an engine. The stress distribution is very uniform. The reduction of the surface pressure at the gap, *i.e.*, the collapse, is not apparent. The sum of all stresses on the side opposite the gap is somewhat greater than those occurring on the gap side of the ring. The cause of this is the error in measurement which has already been fully discussed. This error is a characteristic of the measuring method.

Figs. 7 and 8 have been obtained from two rings both of which have been run for some 2000 km. Both rings were heat formed. The first ring had a uniform pressure distribution while the second had a high pressure point at the gap. The stresses are partly

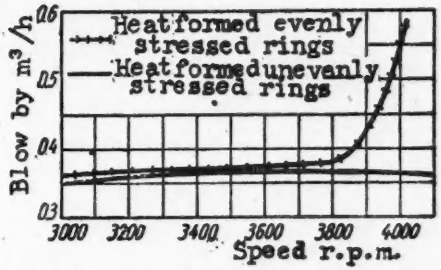


Fig. 9. Gas blow by with heat formed rings having uniform pressure distribution and high pressure point or uneven pressure distribution.

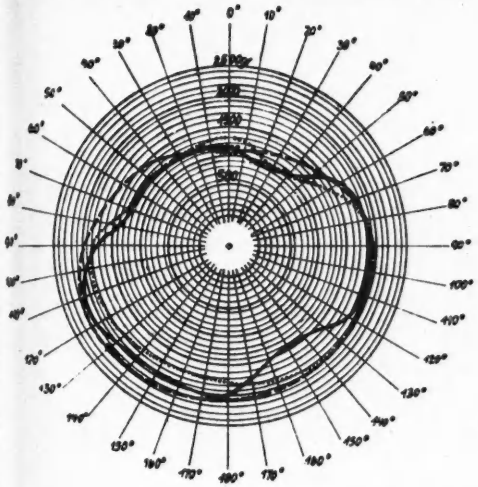


Fig. 7. Heat formed rings from an Opel engine after 2,000 km., with evenly distributed pressure. Mean specific pressure obtained from the tangential force to close the ring, pm = 1.28 kg/cm².

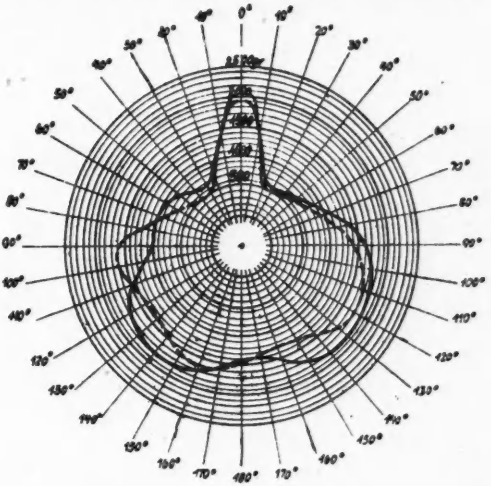


Fig. 10. Heat formed high pressure point ring with exaggerated high pressure point near the gap and sections of minimum pressure next to it. New rings.

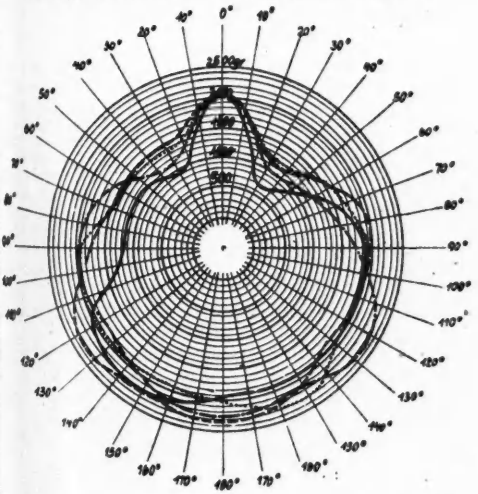


Fig. 8. Heat formed rings from an Opel engine after 2,000 km., with high pressure point. Mean specific pressure determined from the tangential force to close the ring, pm = 1.35 kg/cm².

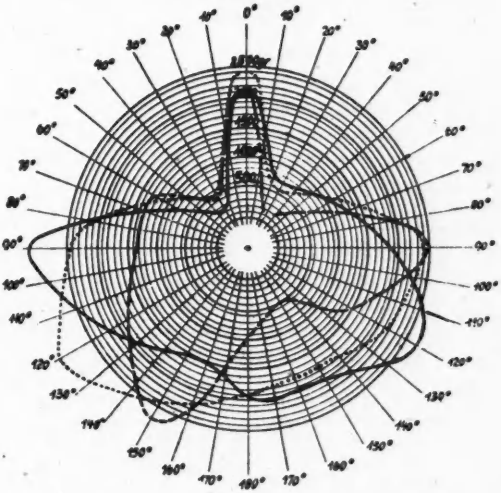


Fig. 11. Turned ring with exaggerated pressure point at gap, followed by points of minimum pressure at other sections of the ring circumference. Mean specific pressure determined from the tangential force to close the ring, pm = 1.42 kg/cm².

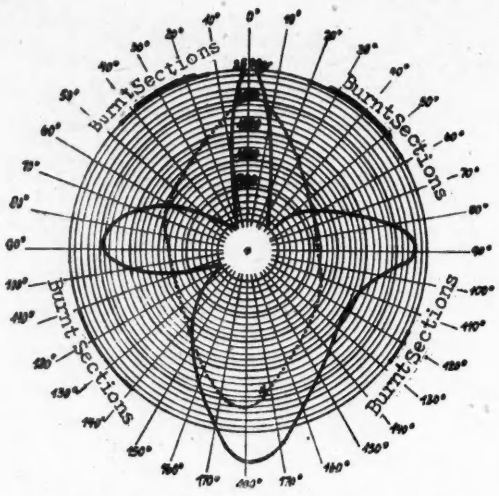


Fig. 12. Turned high pressure point ring with burnt sections. The radial pressure distribution is exaggerated at the gap and is useless. The stress determined by the steel band method is given by the broken line diagram. Time of running not known.

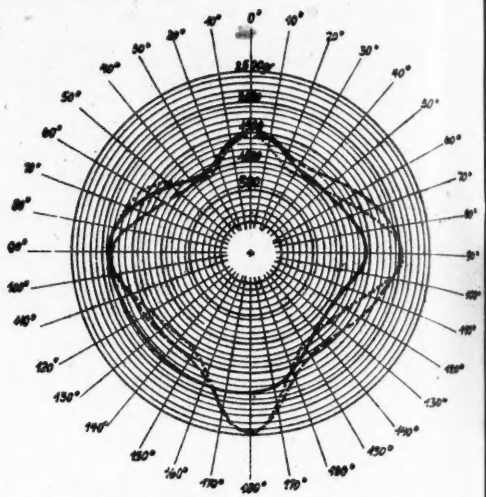


Fig. 13. Turned high pressure point ring with determined pressure distribution. No exaggerated high pressure point at gap and absence of low pressure points in the surface pressure. Mean specific pressure determined from the tangential force to close the ring, $pm = 1.35$ kg/cm². New rings.

typical. The toledo stress of both rings are in agreement.

It can be seen that high pressure points can be obtained with heat formed rings. It is interesting to note that, while with a ring having a uniform ring pressure distribution, the piston blow-by increased above 3800 r.p.m. (Fig. 9), the blow-by remained practically constant with a heat formed ring which had uneven pressure distribution. The opinion that the phenomenon of ring flutter can be overcome by the use of rings with high pressure points, is confirmed by the above measurements.

Attention should be paid by the manufacturers that high pressure points at the gap are not exaggerated, otherwise rings are produced having ring pressure characteristics as shown in Fig. 10. The high pressure at the gap is exaggerated and directly next to it a position of very low pressure is shown, so that the ring can hardly bear against the cylinder. The result is a break through of the hot combustion gases at the dangerous section and excessive heating of the rings, just at its weakest point. In an engine of low output the progressive running-in can remove at first the position of high pressure and eliminate the position of low pressure. In the high duty engine, the ring is often disturbed at the point of its minimum stress due to overloading of thermal stresses, before the running-in process can be of assistance.

This type of unsatisfactory stress distribution is especially found in out-of-round rings. Fig. 11 shows a series of radial pressure distribution curves of piston rings, which have been picked out from a batch of rings.

A number of these out-of-round rings used in a high duty engine had to be replaced after short running periods, because at different parts of the circumference the rings showed burnt sections. The measurements of the rings showed ring pressure distribution curves as shown in Fig. 12. The position of the burnt sections occurring at the circumference of the ring are indicated and prove that the measurements of pressure distribution are correct. Further measurements obtained, with the steel band method, are shown in Fig. 12 by the pressure curve indicated by the broken line. This shows that the ring ovality is about 6 per cent., but does not show the characteristic of minimum ring pressure which was apparent as indicated by the burnt or discoloured ring sections. That out-of-round rings with satisfactory ring pressure distribution can be manufactured is shown by the curves in Fig. 13. This type of ring, formed by turning, is especially of interest in that it was manufactured from plain cast-iron. Its acknowledgeable quality is not due to highly developed materials, but apparently due to careful balancing of the ring pressure distribution, which was very uniformly maintained with all rings. The ring stress at the gap is not very great, but the points of minimum pressure which occur next to the gap are hardly noticeable.

SMALLER TRANSFORMERS BY FORCED COOLING

By W. G. James, Westinghouse Electric & Manufacturing Co. (From *Westinghouse Engineer*, Vol. 2, No. 3, August, 1942, pp. 97/99)

Circulation of the oil of a transformer through the windings and the radiators by a motor driven pump reduces the amount of oil required and the temperature gradients between hot spots in the winding and the cooling oil. The result is that a transformer of given capacity is from one-third to one-half lighter in weight and correspondingly smaller in dimensions. The reduced bulk means that transformers of much larger rating can be shipped completely assembled. A further advantage is that the coils can be placed horizontally. This arrangement often increases the adaptability of the transformer to specific requirements.

Forced-oil cooling reduces for normal 60 cycle designs the steel core about one-half; the copper and insulation a fourth; the oil over a half; and the steel for parts nearly two-thirds. For example, in the case of a

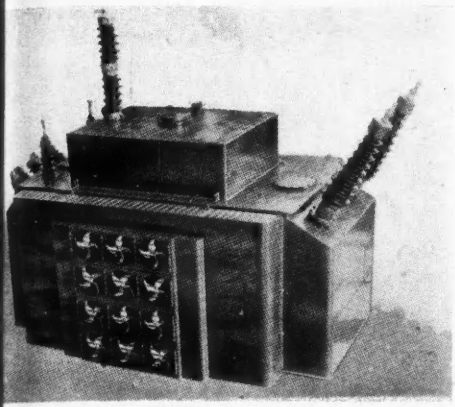


Fig. 1. 40,000 kVA in space requirements of 27,000 kVA self-cooled unit.

40,000-kVA transformer, the steel core can be reduced by 24,000 pounds, the copper and insulation by 3,800, oil by 27,000 and structural steel by 42,000 pounds. The overall weight saving is about 97,000 pounds or 45% of the total. The physical size of the 40,000-kVA forced-oil-cooled, air-blast transformer is about equivalent to a 27,000-kVA self-cooled unit (Fig. 1).

Applicability of the New System.

The new type of transformers had its beginning with several built for electric locomotives. Single-phase transformers, 4,850-kVA, 25 cycles could be placed into a space 7½ ft. by 5½ ft. by 7 ft.

The forced-oil-cooled air-blast transformer is also ideally suited to mobile units such as emergency power service (Fig. 2). Many of these units are in service, the largest rated at 3,000-kVA, 33 to 2.5-kV.

For transformers of larger sizes for usual power systems the principle of forced-oil, forced-air-cooling has the advantage, that much larger units can be shipped from factory to site completely assembled and processed, ready for installation. Several 40,000-kVA. high voltage units (Fig. 3) have been shipped complete with bushings and oil, which only required lifting to their foundations and making the electrical connections in order to place them in service. This saves much time and reduces erection costs considerably. In order to achieve a low overall height, the coils are placed horizontally and the high-voltage bushings are placed sometimes at a lower level than usual (Fig. 1).

A forced-oil and forced-air-blast transformer may be used in the following way. The unit may be operated as self-cooled transformer at normal 40° C. ambient up to a certain load, e.g., 6,000-kVA., fans and pumps being idle. When the load rises above 6,000-kVA., a thermostatic relay in the transformer oil automatically causes the fans to start. The transformer then operates as a self-cooled, air-blast unit with an increase in rating of 33%, or to 8,000-kVA. On an increase in load beyond 8,000-kVA. a second pair of relay contacts closes automatically to start the oil-pump. This increases the capacity by another 33%, or to 10,000-kVA. For operation beyond 10,000-kVA., the usual copper temperature relay takes charge to give a warning signal, when a dangerous temperature is approached and removes the transformer from the circuit, if the temperature rise should exceed safe operating limits.

The amount of materials used in a transformer can be further reduced by replacing the radiators of usual

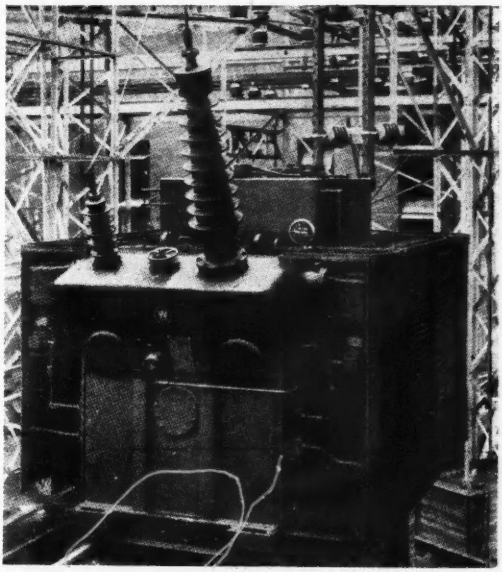
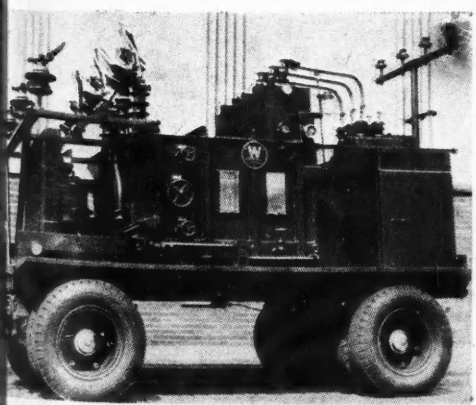


Fig. 3. This 40,000 kVA, forced-oil-cooled transformer was shipped as a single unit.

design by a water cooled heat exchanger. The transformer and its oil pumping system remain the same as before, except that the hot oil is forced through a heat exchanger, which transfers the heat to circulating water. The heat exchanger has to be placed so, that if any leak occurs in the heat exchanger, oil will leak into the water instead of water into the oil. The saving in weight is considerable, as much as 10,300 pounds in the case of a 40,000-kVA. transformer.

Forced Cooling Entails no Radical Construction Change.

The application of forced-oil, forced-air-cooling does not involve any fundamental construction changes. Neither changes in the core and coils nor changes in the intercoil cooling-oil duct system are required. The banked radiators are attached to the tank in the conventional manner, except that the pump is inserted in the oil pipe between the bottom of the radiators and the tank inlet (Fig. 4, 5 and 6). The radiators can be either



g. 2. Mobile power ready for any emergency, any place.

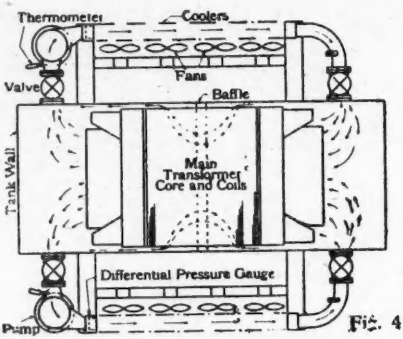


Fig. 4

Forced circulation cooling system of 40,000 kVA transformer.

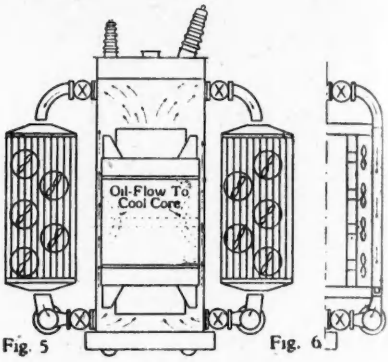


Fig. 5 Fig. 6

Fig. 5. Forced-cooling system with conventional radiators. Fig. 6. Spiral-fin radiators also used in forced-cooling systems.

of the usual flat, expanded-sheet-metal type, or of the newer form, in which thin spiral-fins are tinned on to heavy-wall seamless tubing. The new type is the more efficient radiator, when operated in conjunction with forced air.

The air-blast fans are used individually or in bank assemblies of three or four, driven by totally enclosed single-phase, ball-bearing motors consuming about 150 watts each. The fans use bird-wing impellers for quietness and have a free air delivery of about 2,500 cubic feet per minute.

The pumps are of the centrifugal type driven by 5-kW. 440 volt single phase motors. When auxiliary power is not available, they can be supplied from auxiliary transformers located inside the main tank. The pumps have especially large throats, so that the impellers impose practically no resistance to the thermo-syphon flow of oil, when the transformer is operating without the pumps running. The windings of the motor are submerged in the transformer oil.

The Effectiveness of Forced-oil Cooling.

In the normal self-cooled shell-form transformer the oil rises through the straight vertical ducts by natural convection at about eight feet per minute. The temperature difference of the oil at the top and the oil at the bottom is about 15° to 20° C. With rapid circulation of oil (8 to 10 times the thermo-syphon rate) the temperature difference becomes 2° to 3° C., a marked reduction of the above figures.

When oil flows slowly upwards through the ducts a relatively stationary layer of hot oil forms along the coil surface, which impedes the heat flow throughout the film to the oil. For this reason it is not unusual to obtain a thermal drop of approximately 10° C. from copper to oil—two degrees drop through the insulation and eight degrees through the oil film at the coil surface. Oil forced at a high velocity through the ducts (8 to 20 times the thermo-syphon speed) creates a scouring action or turbulence at the coil surface so that the surface heat drop is reduced to about three degrees or less. The gradient from copper to oil then becomes about five degrees, or only half that of the thermo-syphon gradient. Thus for forced-cooled units, the watts per square inch of coil surface can be materially increased and the amount of copper greatly reduced, without exceeding the thermo-syphon gradient.

For the forced-oil, forced-air cooling, the hottest spot winding temperature is about 8° C. less than for a thermo-syphon unit of the same continuous capacity. That is, a 55° C. self-cooled unit will have a hottest spot winding temperature of 105° C., when operating in an ambient of 40° C. The forced-cooled unit operating under the same conditions, with the same top oil temperature, will have a hottest spot winding temperature of 97° or 98° C. No greater loss of life would result, if full advantage were taken of this condition and forced-oil, forced-air units were rated on a basis of 62 or 63 degrees average copper temperature rise instead of 55° C. as the hottest spot winding temperature under this condition would not exceed 105° C., which is permissible for class A insulation.

THE COMBI-MOTOR OF THE MASCHINENFABRIK ORLIKON

(Abstracted from the *Orlikon Bulletin* in the *Schweizerische Bauzeitung*, Vol. 120, No. 11, September 12th, 1942, pp. 130/131).

THE three-phase squirrel-cage motor is one of the simplest design, but it has two disadvantages: (*a*) the high starting current, which limits the use of such motors although the starting current may be lowered by some special devices, such as by star-delta switches or by starting transformers, and (*b*) the low starting torque. These disadvantages may be reduced by increasing the resistance of the rotor winding. But this works out unfavourably in the later phase of starting, as then the higher resistance of the rotor reduces the torque. The lower the resistance of the rotor winding, the nearer the final speed approaches the synchronous speed and the higher the efficiency of the motor at full speed. Therefore, motors for higher output or for higher starting torque are provided with a starting resistance which is cut out gradually during the starting period. A motor with centrifugal starter has been developed in order to avoid the manual resistance switching. The centrifugal force performs the gradual cutting out of the starting resistance built into the rotor in this design. Recently, the Maschinenfabrik Örlikon has designed the Combi-Motor, the rotor of which also contains an automatic starter.

The rotor of this model has two windings, I and II (Fig. 1). At first no notice should be taken of the winding I. If the contact K_1 of the then only winding

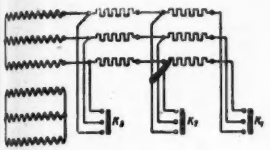


Fig. 1. Diagram of the motor.

II would be permanently closed (starpoint), the winding II would behave like the winding of a rotor with a centrifugal starter with two starting steps of usual design. When the motor is switched in and speeds up, a part of the starting resistance built into the rotor would become short-circuited by the centrifugal switch K_2 and then, at a higher speed, the remainder by the switch K_3. Naturally a decrease in speed, e.g., in consequence of overload, interruption of one phase, decrease of voltage or frequency would have the effect that the contact K_3 or even the contact K_2 would open once more. It would be necessary to provide an automatic device (centrifugal switch with relays) for the protection of the starting resistance in winding II, which is unsuitable for continuous loading. Such a special automatic device is unnecessary for the Combi-Motor.

In this motor not only K_2 and K_3 are centrifugal switches, but also the starpoint K_1, and a special patented design secures not only the closing but also the opening of these three switches in the order: K_1, K_2, K_3. This has the consequence that the starting resistance is open and does not carry any current when after a regular starting the speed decreases again for some reason. Switching in of the motor after a shut down would find the centrifugal switches open, so that no current would be produced in the winding II. Therefore, a second squirrel cage winding, I, is provided for starting the motor. The resistance of this winding may be such that a low starting current and a high starting torque results. Such resistance could not be applied in a normal squirrel cage motor for the reasons already mentioned. These reasons have no influence on the choice of the resistance of the winding I,

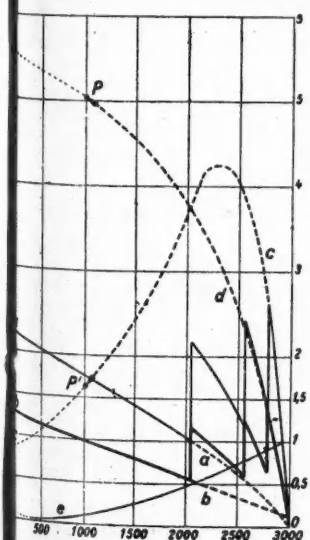
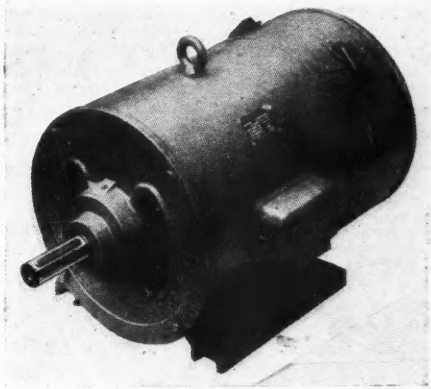


Fig. 2

Starting of a two pole combi-motor driving a centrifugal pump.

(a) torque and (b) starting current with winding II open; (c) torque and; (d) starting current with short-circuited winding II; (e) load; (p) P current and torque which make contact K open.



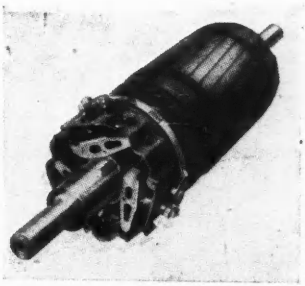


Fig. 3

(Top) Combi-motor with rotor (Left) The cover has been removed from the centrifugal starter.

because in the second phase of the starting, that is when K_1, K_2 and K_3 have closed successively, the winding II is in parallel with the winding I and the resistance of the winding II, which is short-circuited at full, normal speed, may be made as low as required.

The starting process of a motor of this design is shown in Fig. 2, the machine is supposed to have a load curve according to curve (*e*). In the first phase squirrel cage winding I alone is working. Curve (*b*) shows the current and curve (*a*) the driving torque. The second phase begins, when the speed of the motor has attained the value at which the contact K_1 is closed, that is with the motor described at about two-thirds of the synchronous speed. Then the insulated winding II is short circuited across the highest starting resistance. The reduction of the resistance of the rotor, now working with both windings, produces an increase of the current as well as of the torque at this speed. A similar effect is produced by the subsequent second and third lowering of the resistance of the winding II when the switches K_2 and K_3 are closed automatically. The final current and torque are demonstrated by curves (*d*) and (*c*). The intersection between the curves (*e*) and (*c*) gives the normal working point of the motor and the maximum of curve (*c*) corresponds to the pull-out-torque. If in consequence of a sudden increase of the load the current assumes a very high value and the motor pulls out, the thermal fuse disconnects the motor. It has been mentioned, that a disturbance in the supply system may also cause a decrease of speed. The increase of the current, which is a consequence of the decrease of speed, is stopped by the interruption of winding II. The disconnection of the motor by its cut-out would be prevented by a premature interruption of the winding. Therefore, the speed, at which the contact

K_3 opens as the last of the three contacts, is set very low.

In the centrifugal starter of the Combi-Motor (Fig. 3), the closing and the opening of the toggle-lever switches K_1, K_2 and K_3 is performed by the centrifugal force counteracting the force of a spring pressure. The lever, on which the spring is acting, is shortened and in consequence its moment with respect to the axis of the switch is lowered by the closing operation of the switch. Therefore, the moment is greater in the open position of the switch than in the closed position: $Do > Dz$. The switch closes as soon as the moment Z of the centrifugal force exceeds Do at the closing speed n_s:

$$Z(n_s) \geqslant Do \quad .. \quad .. \quad (1)$$

The contact is reopened, when during a retardation of the motor to the speed n_r (re-opening speed) the closing moment of the centrifugal force Z has fallen below the value of the force exerted by the spring—

$$Z(n_r) \leqslant Dz \quad .. \quad .. \quad (2)$$

Therefore,

$$Z(n_r) < Z(n_s)$$

and

$$n_r < n_s.$$

A stronger spring raises Do and according to (1), the speed n_s at which the centrifugal switch is closed. The shorter lever arm upon which the spring acts when the contact has been closed, lowers Dz and in accordance with (2), the speed n_r. By appropriate mechanical and geometrical dimensioning of the switches, it is not difficult to arrange that the contact K_3 closes and opens last.

It can be seen from what has been said, that the Combi-Motor with its automatic starting device is suitable for drives with remote control and for machines with varying load, and in general, where the use of a simple squirrel-cage motor or a motor with double squirrel-cage rotor are not permitted by the supply company because of the starting current.

FELT MEETS WARTIME DEMANDS

By COLIN CARMICHAEL. (From *Machine Design*, Vol. 14, No. 10, October, 1942, pp. 65-68.)

THE war has greatly accelerated developments in the use of felt as an engineering material. Applications of current importance include temperature and sound insulation in aeroplane cabins, vibration isolation of production machines and within army tanks, oil and grease retention in aero-engines, lubrication of tools, etc. The wide range of functions is possible because this type of fabric can be manufactured in an extensive series of grades, producing pre-selected combinations of physical properties.

The felt-making process is made possible by the tendency of certain fibres to mat or knit together when subjected to heat, moisture and kneading or beating. Wool has the tendency to "mat" to a far greater degree than any other fibre. Under the microscope wool fibres have the appearance shown in Fig. 1.

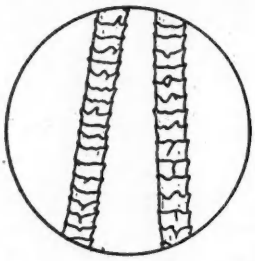


Fig. 1

The projecting edges of the fine overlapping scales form serrations which lock the fibres together during the felting process. Other animal fibres such as hair have relatively few serrations, while vegetable fibres have none and cannot form felt without the presence of wool.

Control of the grade and percentage of wool, and the duration of the kneading or beating operation, provides a wide variety of classifications. By standardizing on relatively few of them, the felt industry has eliminated needless variations, and has eased the task of the designer in his selection. Those of industrial importance are listed in Table 1.

In general, back check felt is used where high strength and hardness are needed. If the felt is not compressed, it is suitable for such applications as oil retention. Extra firm pad felt is particularly resilient. Pad felts F-10, F-11 and F-12 are recommended for grease and oil retention where the felt is confined and compressed in assembly. F-13 and F-15 are suitable for sound deadening. Soft pad felt can be used only when held in place between other materials.

Table 1. *S.A.E. and Felt Association Standard Felts*

S.A.E. No.	Felt Assoc. Designation	Trade Classification	Chemical Analysis% Wool (min.)	Chemical Analysis% Ash (max.)	Spec. Gravity	Breaking Strength (psi)	Oil Absorption* (%)
F-1	B-30	Back Check	95	1.0	.342	500	215
F-2	B-35	Back Check	95	1.5	.342	500	230
F-3	B-40	Back Check	90	1.5	.333	400	190
F-5	C-30, C-35	Extra Firm Pad	95	1.0	.261	400	—
F-6	C-40, C-45	Extra Firm Pad	92	1.5	.261	275	370
F-7	C-50, C-55	Extra Firm Pad	80	2.0	.261	250	290
F-10	D-10	Firm Pad	95	1.0	.182	225	450
F-11	D-20	Firm Pad	92	2.0	.182	200	475
F-12	D-30	Firm Pad	85	2.0	.182	100	500
F-13	D-40	Firm Pad	75	2.5	.182	75	525
F-15	D-50	Firm Pad	55	4.0	.182	75	500
F-26	D-56	Soft Pad	45	5.0	.162	—	625
—	D-70	Kapok	—	—	.085	—	—
F-50	H-10	Ball Bearing	95	1.0	.320	500	225
F-51	G-10	Ball Bearing	92	1.5	.320	300	—
F-55	G-30	Lining	75	3.0	.245	200	—

* Percentage increase in weight when impregnated in oil at 76° F. Values are average for S.A.E. oils 10 to 50.

Kapok felt has been developed for heat insulation and sound absorption. (Composition: ca. 45% kapok, 30% cotton, 25% wool.) By suitable treatment it can be rendered flameproof and mothproof. Thermal conductivity is 0.21 B.Th.U. per hour per sq. ft. per degree Fahr. per inch thickness. Because of the kapok fibres, moisture absorption is low. Sound-absorption coefficients, which measure the fraction of sound energy which disappears at a single reflection from the surface, are given in Table 2.

Ball-bearing felts are thin smooth felt used for oil retainer washers, gaskets, etc. Lining felt is recommended for anti-squeak strips and for lining when cemented to metal panels.

Modification of the physical properties of many of these standard felts is possible by special treatments.

Table 2. Sound Absorption Coefficients of Kapok Felt.

Thickness (Inches)	Coefficients for various frequencies 128 (cps†)	256 (cps)	512 (cps)	1024 (cps)	2048 (cps)
⅛	.16	.21	.30	.53	.58
¼	.17	.24	.40	.65	.74
⅜	.18	.27	.45	.70	.77
½	.19	.29	.51	.75	.80
¾	.20	.31	.62	.81	.83

† Frequencies in cycles per second.

Significant developments are taking place in the use of felt impregnated with plastics and rubber.

Colloidal graphite impregnation gives a material specially suited for lubrication purposes. Felt has countless uses as an aid to lubrication, be it as wicking or as a saturated reservoir that will hold a large amount of oil. Oil-absorbing ability and ash-content are listed in Table 1.

Versatility of felt is well illustrated by its use as filters in industrial instruments, for sealing purposes to prevent escape of lubricant and to exclude dust from bearings. Felt is used also as wipers for machine ways, keeping dirt and chips from the rubbing surfaces.

As a machine mounting for the isolation of vibration felt has marked advantages. Considerably more compressible than rubber, it does not require freedom for

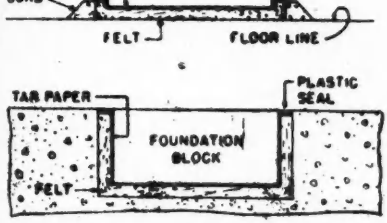


Fig. 2

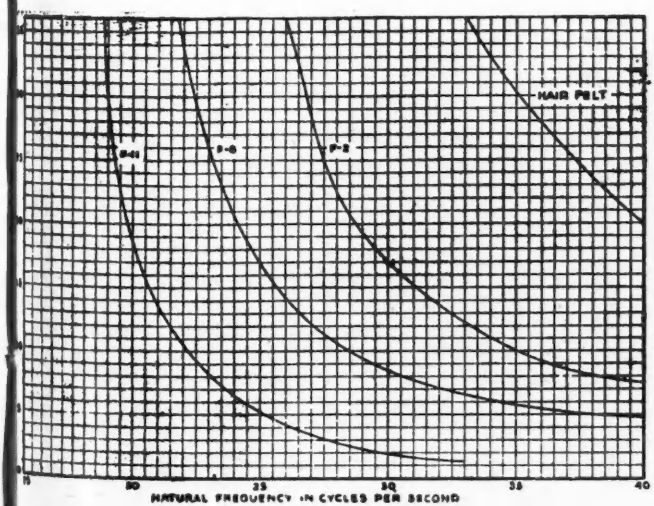


Fig. 3

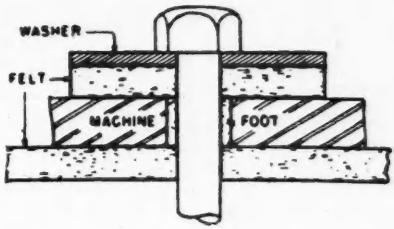


Fig. 4

lateral expansion and hence can be used in the form of sheets or pads of relatively large area compared with its thickness (Fig. 2), as well as in the form of washers (Fig. 4).

When designing a felt mounting for vibration isolation, basic principles of mechanics may be applied. In reducing the force transmitted to the foundation by a periodic disturbing force on the machine, the ratio of amplitudes of the transmitted to the disturbing force—called the transmissibility—is expressed by the equation

$$\epsilon = \frac{1}{r^2 - 1} \quad .. \quad .. \quad .. \quad .. \quad (1)$$

where ϵ = Transmissibility,

$$r = \frac{\text{Disturbing frequency}}{\text{Natural frequency}} = \frac{f}{fn} \quad .. \quad (2)$$

Effective isolation requires $r > 1.414$; in general, $r = 2$ to 3 gives satisfactory isolation.

The natural frequency of a machine on one elastic support is given by the equation

$$fn = \frac{1}{2\pi}\sqrt{\frac{kg}{W}} = \frac{1}{2\pi}\sqrt{\frac{g}{s}} \quad .. \quad .. \quad (3)$$

where

k = stiffness or spring rate of the support (lb./in.).
W = weight of the machine (lb.).
g = acceleration due to gravity (386 in./sec.²).
s = static deflection of support due to W (in.).

The static deflection may be written

$$s = \frac{Wt}{aE} = \frac{pt}{E} \quad .. \quad .. \quad .. \quad (4)$$

t = thickness of felt (in.).
a = area of felt (in.²).
p = unit pressure on felt (lb./ in.²).
E = elastic modulus of felt (lb./ in.²).

On substitution of (4) in (3), the natural frequency becomes

$$fn = \frac{1}{2\pi}\sqrt{\frac{gE}{pt}} \quad .. \quad (5)$$

Because E varies with p, it is necessary to express this relationship by means of curves, as in Fig. 3. Four felt classifications are shown, each type having a different E. The curves are drawn for t = 1 in. For thicknesses t = n in. multiply the abscissa by $\sqrt{\frac{1}{n}}$

When the disturbing force is small relative to the weight of the machine, felt pads alone are sufficient to hold the machine in place. To counterac tany tendency for the machine to walk away, the mounting pads are to be fixed so as to prevent sidewise motion of the machine, e.g. the felt pad can be glued to both foundation and machine with a special adhesive which

can be softened with a suitable solvent whenever it becomes necessary to move the machine.

In cases where large disturbing forces call for anchor bolts to hold the machine down, special care is necessary to prevent the conduction of vibrating forces through the bolts themselves. A suitable design using a combination of felt washers and pads is shown in Fig. 4.

For heavy machinery such as drop hammers, punch presses, etc., the entire concrete foundation block under the base is poured on tar paper laid over a felt blanket (Fig. 2).

In considering possible uses of felt in machines, designers will do well to bear in mind the wide range of possible properties that can be obtained—some felts being hard enough to turn on a lathe, others soft enough to be sewn.

The illustrations included in this article were supplied by: The Booth Felt Co. (Fig. 1).
The American Felt Co. (Fig. 3).
The Felt Association (Figs. 2 and 4).

AUTOMATIC ELECTRICAL APPARATUS FOR MEASURING THE MOISTURE OF COAL

(From *Elektrotechnischer Anzeiger*, Vol. 59, No. 10, 13th May, 1942, pp. 221-222.)

THE mechanical strength and the durability of briquettes depend to a high extent on the moisture content of the brown coal. In order to produce briquettes of uniform good quality the content of water of the coal must not differ by more than 2% from the optimum. Various methods have been devised for the uninterrupted measurement of the moisture content of the ground coal. The method most frequently used up to now employs a drying chamber, in which samples of the coal, the weight of which is exactly measured, are dried by heating to approximately 100° C. and by applying a current of warm air. The loss of weight corresponds to the original moisture content.

This method is very accurate, but takes too much time. It would not be advisable to shorten the time by raising the temperature, as volatile parts of the coal may evaporate.

As in general the briquette-press works continually, the moisture, too, has to be measured without interruption. With a hand-operated arrangement this is tiresome and it is difficult nowadays to find the skilled labour.

The A.E.G. has succeeded, after endeavours lasting many years, to overcome many difficulties, and to design, in collaboration with the coalmine in Braunschweig (Germany), an instrument which measures automatically the moisture content, and which has been found suitable for all practical working conditions.

This apparatus works according to the drying chamber principle. The time for the drying has been shortened by drying the coal in a drum (Fig. 1) of special design. As long as the drum turns in the direction of the slots, the coal particles are kept together in the drum, in a constant whirl, and are dried in the shortest possible time by a warm air current streaming through the drum. After some time, ascertained empirically, the direction of rotation of the drum is reversed. This makes the coal fall out of the drum on to the pan of a balance. Even the smallest particles leave the drum.

An advantageous feature of the new instrument is that it is not necessary to use a coal sample of a certain weight. The weight of the sample is ascertained before and after drying on the same balance, and an electric measuring instrument, coupled to the balance, registers the loss of weight of the sample as a percentage of the original weight. This is indicated by the electric current adjusted through the deflection of the balance. The indicating instrument may be fixed at a convenient place, and there is no objection to connecting up several instruments with the same indicator. The current produced may even be used for influencing the manufacturing process, e.g. for regulating the drying plant of the briquett press, in order to produce a material of constant moisture.

The arrangement of the apparatus is shown in

Fig. 1. Drying-drum of the automatic A E G Moisture Tester.

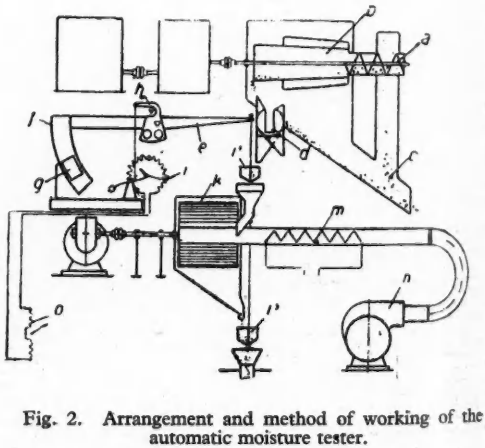


Fig. 2. Arrangement and method of working of the automatic moisture tester.

Fig. 2. The sampling is made by the worm gear (a). The coal passes the sieve (b) and the material of the correct and uniform size is collected in (d). When this vessel is full, it is emptied into the upper pan 1 of the balance. The deflection of the beam of the balance is transmitted to the sliding contact (i) and the corresponding resistance is registered. In order to attain quick damping of the beam, it swings between two permanent magnets (g). As soon as the weight is registered, the coal is emptied into the drum (k), the function of which has been described above. The dried coal falls into the pan 1 and is weighed again. The resistance, which corresponds to the deflection of the balance, is registered and the measuring instrument indicates the content of water in the sample. The pan is emptied and a new sample is poured into the upper pan.

One measurement takes 7½ minutes; 1½ minutes are taken by the various mechanical processes and 6 minutes by the drying process.

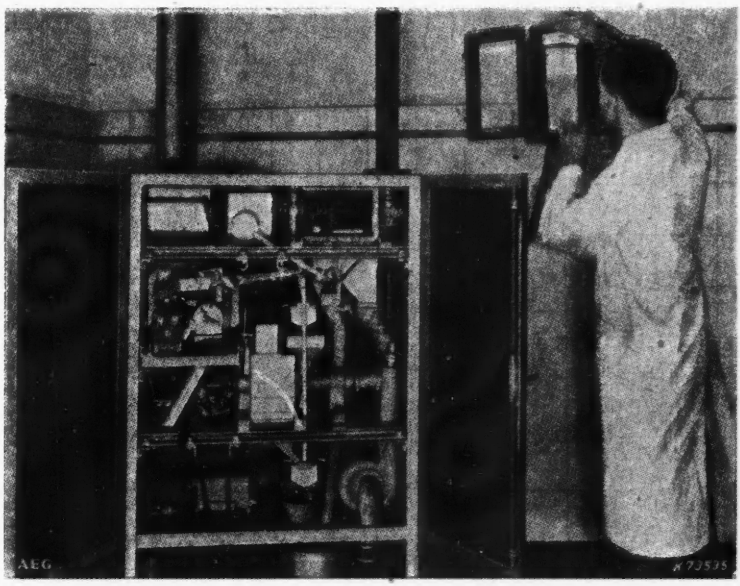


Fig. 3. A E G Moisture Tester for coal powder with automatic recording instrument.

The complete apparatus is shown in Fig. 3. The steel casing has a height of 128 cm., a width of 80 cm. and a depth of 50 cm. The cross-coil recording instrument is fixed on to the wall.

SUPER-CHARGING OF TWO-STROKE DIESEL ENGINES

By F. Oederlin, Managing Director, Sulzer Brothers Ltd., Winterthur. (From *Swiss Technics*, Vol. 21, No. 3, August, 1942, pp. 12-17.)

Supercharging of two-stroke Diesel engines has been one of the most important problems of the development of Diesel engines. Already in 1912 Sulzer two-stroke engines were built with "extra charging." The extra charging pressure was limited to about 17-20 lbs./sq. in., giving an additional 10-30% engine power. Until recent years this supercharging pressure remained unaltered as further increasing the pressure would have raised the fuel consumption. The utilization of the energy of the exhaust gases has opened the way for new development of supercharging.

The simplest solution for driving the supercharger, or blower, would be the employment of an exhaust gas driven turbine. Unfortunately, it is not possible to realise this method in practice, since only at high loads and high exhaust gas temperature is the output of the exhaust gas turbine sufficient for driving the blower. An even greater defect of this system would be that the engine could not be started, as the exhaust gas turbine reaches normal working speed too slowly. Consequently, to supercharge two-stroke engines, the energy necessary to drive the blower must be supplied from an ambient source.

If, however, the scavenging and supercharging pump is driven from the main shaft, and the exhaust gas turbine is transmitting its power to the main shaft through gearing, then it is possible to employ high supercharging pressures, even during starting. Theoretically, the supercharging pressure can be regulated at will, and for highest efficiency it should be adjusted according to speed and load.

Investigations have shown that the indicated output of the exhaust gas turbine is greater than the indicated power absorbed by the compressor. A positive area of work is, therefore, available for the charging set. However, because of the unavoidable losses in the exhaust gas turbine and compressor, the output of the turbine is sufficient to cover the power required by the compressor only in the case of large installations and when the load on the plant is high. The surplus power of the supercharging set is then transmitted to the crankshaft through a gearing. But even with engines of medium and small output the power required by the compressor will be supplied mostly by the exhaust gas turbine. Under average conditions it can be presumed that the power required for the compressor and the output developed by the turbine will balance one another to a large extent.

Based on these considerations, Sulzer Brothers took up, several years ago, the practical work on supercharging two-stroke engines. First of all a special experimental engine was built and submitted to thorough testing. These tests were at first confined to the engine alone. The influence of the exhaust gas turbine was limited by a throttle valve. The air for scavenging and charging was taken from the compressed air system in

the works and heated up to the temperature corresponding to the polytropic compression of a normal compressor.

With the experimental engine supercharged to 2 atm. a mean effective pressure of 170 lbs./sq. in. was attained with absolutely clear exhaust. This mean effective pressure could be maintained without any difficulty for a considerable length of time. Compared with the non-supercharged two-stroke engine, the increase in mean effective pressure or in output was 100%.

The same engine was then worked with a supercharging pressure of 3 atm. The corresponding mean effective pressure was 210 lbs./sq. in. The supercharging pressure could be gradually increased to 6 atm. and the mean effective pressure to 255 lbs./sq. in. Combustion takes place in a relatively large proportion of excess air, and gives at all loads an invisible exhaust gas most suitable for driving an exhaust gas turbine.

Based on the excellent results obtained with the experimental engine, an engine supercharged to 2 atm. was built. The principal data of this opposed-piston engine are as follow :—

No. of cylinders	4
Bore	190 mm.
Stroke	2 × 300 mm.
Speed	750 r.p.m.
Piston speed	7.5 m./sec.
	1,475 ft./min.
Power	1,370 B.H.P.
M.E.P.	12 kg./cm.².
	170 lbs./sq. in.
Fuel consumption	0.35 lb./B.H.P. hour.

During industrial service in the Sulzer works, for more than 3,000 hours, it was found that the engine complies in all respects with the stipulated conditions. The mechanical transmission of energy between exhaust turbine and crankshaft never gave rise to any trouble. Even during periods of severe cold, the engine started easily and quickly.

In large installations the mechanical coupling between the turbine and the crankshaft can be replaced by an electro-magnetic or a hydraulic coupling.

In order to determine the effect of supercharging at higher speeds an experimental engine was built with cylinders of 120 mm. bore and 2 × 150 mm. stroke. It was first run at 1,500 r.p.m. Supercharging pressure and mean effective pressure was the same as before, and the fuel consumption was 180-190 grams/B.H.P. hour.

In addition tests were made on a single-piston engine of 420 mm. bore. Supercharging was limited to 2 atm., and it was found that up to this limit the output increases proportionally to the amount of supercharging. Sulzer Brothers have, therefore, started to redesign their single-piston engines of medium and large bore according to the requirements of two-stroke supercharging. Simultaneously, they have developed a new type of engine (Fig. 1), allowing far-reaching utilization of the possibilities offered by supercharging.

Supercharging to 5 or 6 atm. pressure represents a unique case, in that, within this range of supercharging, the power delivered by the Diesel engine and the power absorbed by the supercharging compressor become equal. The effective output of the whole plant thus corresponds substantially to the output of the exhaust gas turbine. The turbine may be therefore

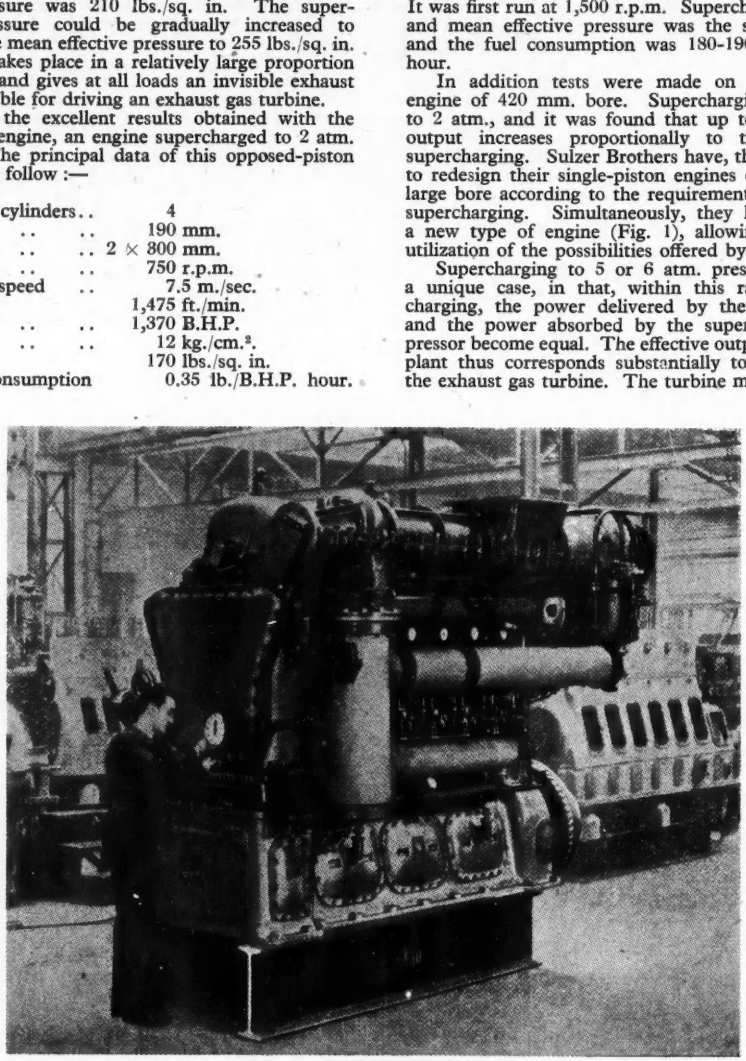


Fig. 1. Two-stroke two-shaft opposed-piston Diesel engine supercharged to 2 atm. abs. and developing 1560 B.H.P. at 850 r.p.m. in the Sulzer works at Winterthur.

uncoupled from the Diesel engine and from the compressor without the energy balance of the whole set being thereby disturbed. The set comprising Diesel engine and compressor, here designated "power gas generator," fulfils the same purpose as, for instance, a boiler in a steam power station. The whole effective output is given up by the power gas turbine, which corresponds to the steam turbine of the steam power station. Accordingly, several power gas generators can be arranged to work on one common power gas turbine. This power generating process can also be accomplished by means of free-piston power gas generators. With power gas generators of this kind, each of the opposed pistons of the Diesel part works directly on a compressor plunger which compresses the scavenging and charging air. No crankshaft is provided. The two pistons are merely coupled to one another by a linkage which ensures a symmetrical operation. The volume of clearance space adjusts itself automatically to suit the instantaneous supercharging pressure.

A particularly remarkable feature is the high thermal efficiency of 35-40% without using any recuperators or similar apparatus, and at service temperatures of only 450 to 500° C. The power gas turbines are small and simple. The reliability in service of the whole plant is increased by the independence of the individual power gas generators. Critical speeds are not to be feared, since the separate power gas generators are coupled to each other only by a very flexible gas column. With the power gas process outputs may be obtained which lay hitherto beyond the range covered by Diesel engine and with specific weights per power corresponding to those of the lightest steam plants. The highly supercharged two-stroke Diesel engine will, in the first place, be employed where strict requirements with regard to weight and space have to be complied with, as, for instance, in traction, in certain kinds of ships, and in general for plants intended for taking peak loads. The supercharging set of the power plant may be regarded as a constant pressure gas turbine, i.e. a gas turbine with continuous combustion whose combustion chamber is replaced by a supercharged Diesel engine. The supercharging set has, as in the case of the gas turbine, an indicated output of positive value. Whilst in the case of the gas turbine only the difference between the output of the turbine and the power absorbed by the compressor appears as effective output, this is increased, with high supercharging, by the output obtained from the Diesel engine. Its effect appears clearly in the thermal efficiency, which is of the order of about 40% with high supercharging, whilst it amounts to about 18% with the simple gas turbine.

Replacing the combustion chamber of the gas turbine by a Diesel engine is also justified with respect to heat considerations since it is in any case necessary to cool the hot gases from their initial temperature of 1,500-2,000° C. to about 500-600° C. before admitting them to the exhaust gas turbine. Instead of effecting this cooling by admixture of excess air, as is the case in the gas turbine, in the supercharged Diesel engine, this cooling takes place mostly through adiabatic expansion of the gases in the cylinders, where they perform useful work at the same time.

If the supercharging pressure of the Diesel engine is further increased, the swept volume becomes smaller and smaller, the clearance remaining the same. In the limiting case the swept volume becomes zero, and the Diesel engine is then reduced to nothing but clearance space. This is then identical with the combustion chamber of the hypothetical high-pressure gas turbine, which, because of its high working temperatures, can at present not yet be realised. On the other hand, this working process can, as has been shown, be carried out with good efficiency in the supercharged engine or in the power gas plant.

From these considerations and from the results obtained, it can be concluded that there are important points in favour of this combination of Diesel engine and gas turbine. It offers an increase in thermal efficiency of more than 100% as compared with the gas turbine, and an increase in mean effective pressure of 100 to 200% compared with the non-supercharged Diesel engine.

Nevertheless the gas turbine should also acquire importance especially in applications where units of large power are needed, provided that its efficiency at full load and particularly at partial load can be considerably improved.

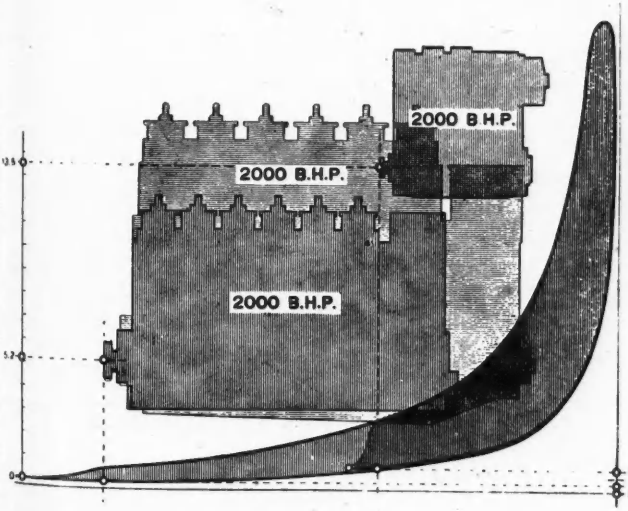


Fig. 2.

Comparison between the space required for a non-supercharged 2000 B.H.P. two-stroke Diesel engine with cross-head, a non-supercharged 2000 B.H.P. trunk-piston two-stroke Diesel engine, and a 2000 B.H.P. two-stroke opposed-piston Diesel engine supercharged according to the process. The darker indicator diagram of the supercharged engine shows the work done in the Diesel cylinders. The work resulting from the exhaust gas turbine and the supercharging compressor is, as regards area, almost identical with the black cross-hatched diagram. Lower diagram area of the non-supercharged engine.

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Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

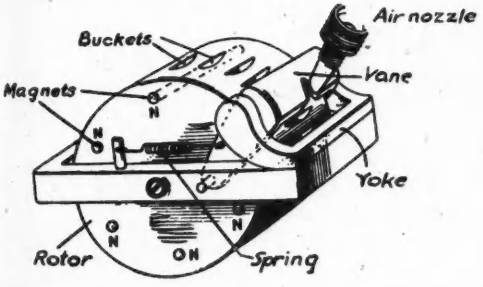
- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

NEW EQUIPMENT

BENDIX GYROSCOPE SPEED REGULATOR.

AIRCRAFT instruments such as rate-of-turn indicators, directional gyroscopes and artificial horizons employ high-speed gyroscopes driven by small air turbines. Air is generally supplied by an engine-driven suction or pressure pump, by using the intake manifold suction or by a venturi tube projecting into the air stream. Pressure variations in the supply line from any of these sources require the use of a governor to keep the gyroscope speed substantially constant. A simple device for so regulating gyroscope speed is the subject of a patent assigned to Bendix Aviation Corporation.

Rotor of the gyroscope is mounted in a gimbal frame and driven by air pressure from a nozzle impinging on buckets in the periphery of the rotor. Control of the quantity of air reaching the buckets is effected by a tapered notch cut in a metal vane pivoted in the frame and normally held away from the nozzle by a spring. Permanent bar magnets embedded in the rotor periodically pass under the vane, and a magnetic circuit is completed through the vane and through an adjacent yoke, which is made of a material having high permeability, such as soft iron. The pivoted vane is copper, aluminium, or other good electrical conductor.



Eddy currents are set up in the vane, due to the moving field, and provide secondary magnetic fields which react with the bar magnet field, causing a drag on the vane which, however, is overcome by the spring tension except when the speed exceeds normal. Increase of speed due to a surge in pressure increases the eddy currents and drag, pulling the vane under the nozzle and reducing the amount of air reaching the buckets. As speed returns to normal, the spring pulls back the vane, permitting full impingement once more.

SPEEDING UP MACHINING OF GEARS.

GENERAL Electric automatic stand-by power equipment has been doing sentinel duty for a number of months at a plant of the De Laval Steam Turbine Co., manufacturers of marine speed-reduction gears. By eliminating time losses due to power failure, and by substantially reducing spoilage in this plant, the equipment has not only speeded the production of these vital gears but has already justified its installation.

Certain of the final hobbing operations in the manufacture of speed-reduction gears cannot be interrupted or even subjected to appreciable speed variation without affecting the accuracy of the work. This is because differences in the momentum of the work with relation to that of the cutting tool would tend to produce variations in the depth of the cut.

To guard against power failure or excessive voltage fluctuation, it was decided to instal a stand-by power source. The primary requirement was that the equipment have a capacity to supply the essential loads, ranging from 17 to 23 kW (75 to 100 amperes at 230 volts), for not less than four hours. Experience showed that power failures would not extend beyond this length of time. Other requirements were : instantaneous and automatic transfer from stand-by to normal source and from normal to stand-by force ; battery charge rates automatically varying from a trickle rate during stand-by periods up to more than normal rate if necessary following emergency discharge periods ; and highest possible dependability.

General Electric engineers working with the Edison Storage Battery Co. designed an automatic power supply system consisting of a motor-generator set, a battery, and control equipment to meet these requirements. The motor-generator consists of one 15-H.P. 230 volt motor and two 5-kW, 50-volt generators. The motor is supplied by the normal source during stand-by periods and by the battery during interruptions in normal source. The generators, each in series with half (96 cells) of the battery, regulate the voltage during both charge and discharge. The control panel is provided with meters, an undervoltage relay, disconnect contactors, and voltage regulators with their control equipment. Two carbon-pile regulators vary the field current of the generators through voltage control across the battery during charge and across the load during discharge. The regulators also perform the triple function of charging the battery at various rates from a trickle to more than normal ; bucking the battery voltage during discharge when above the required load voltage ; and boosting the battery voltage during discharge when it is below the required load voltage.

All regulation is automatic, and is so close that machine operators in the gear hobbing department do not know whether power is coming from the normal source or from the battery.

HARDNESS OF TIN-BASE ALLOYS.

METHODS of obtaining new and stronger tin-base alloys suitable for use as bearing metals is part of the programme of research conducted by the Tin Research Institute.

The effect on hardness, produced by quenching from the highest practicable temperature followed by

prolonged tempering at 100° C. and 140° C., has been examined for 80 tin-base alloys containing 4 to 14 per cent antimony and 0 to 10 per cent cadmium. The results of the hardness tests are recorded in a paper by W. T. Pell-Walpole, B.Sc., Ph.D., in the Journal of the Institute of Metals, Vol. 68, October, 1942.

It is shown that these alloys can be hardened by heat treatment and maintain a useful degree of improvement for at least 1000 hours at 100 to 140° C. The best alloys in this respect are those in the range antimony 9 to 10 per cent, cadmium 1 to 1½ per cent, balance tin. The degree of improvement is indicated by Vickers diamond pyramid hardness tests. Values of 33 to 34 are obtained, compared with values of 26 to 30 in the normal non-heat-treated condition.

Reprints of this paper may be obtained free of charge from the Tin Research Institute, Fraser Road, Greenford, Middlesex.

British Institution of Radio Engineers.—At the November meeting of the British Institution of Radio Engineers, held at the Lecture Hall of the Institution of Structural Engineers, 11, Upper Belgrave Street, London, S.W.1, on Saturday, November 21, Dr. L. E. C. HUGHES read a paper on "The Technical Basis of Sound Reproduction."

After a historical introduction dealing with the definitions of "sound" and "noise," the one wanted and the other unwanted, he described the various degrading effects on sounds by the chamber in which they were emitted or reproduced. The effect of the auditorium on the attributes of sounds: pitch, loudness, acceleration, spatial and contact, and how they were degraded as compared with the original, was explained. He gave a formula for determining the effect in a given auditorium.

The final test was always a subjective one, and owing to the short memory of the ear it was necessary for comparisons between the original performance and the reproduction to be direct. This was difficult, and the method he advocated was to use two separate chambers with a microphone in each to represent the listener, and to compare their outputs subjectively at the remote end of a very good quality amplifier.

He explained that each letter had its own requirements in the matter of frequency range for good reproduction of speech, but he stressed the point that the final test was always an aural one in the tests conducted by the Apparatus Approval Committee of the Central Council for School Broadcasting, where non-technical audiences voted subjectively on the quality of reproduction on a scale attained entirely by experience.

CATALOGUES RECEIVED

Harland Submersible Pumping Set.—A leaflet issued by The Harland Engineering Co. Ltd., Alloa, Scotland, describes in detail the submersible pumping set including pump, motor, cable connections, and switch gear. Against the limitations of these pumps from the point of view of reliability and also efficiency, especially in the larger sizes, for smaller units where an a.c. supply is available, the submersible pump has a definite place in the range of underground pumping equipment. Its main advantages are that it is: simple and inexpensive to instal; power is applied directly to the pump; and high output for a given diameter of well because the unit runs at 2,850 r.p.m. Free copy of the leaflet available on application.

EDITORIAL INDEX

The Editorial Index to Vol. III., January-December, 1942, will be sent to all subscribers free of charge, together with the January, 1943, issue. Binding cases for the binding of Vol. III. are now available. The price is 4/6 including postage.

BUSINESS CHANGES

Compression Units Ltd., 20, Victoria Street, Bristol 1, have moved to new premises at 55, Queen's Road, Bristol 8.

Lindley Duffield & Co. (Metallurgical & Finance) Ltd., Consulting Engineers and Metallurgists, have moved from Imperial House, Kingsway, London, W.C.2, to Cherwell House, Little Tew, Oxford.

Victor Engineering and Sheet Metal Co. have moved to Christie Road, Stretford, Manchester.

CORRECTION.

Explosive Rivets for Metal-clad Control Surfaces.—On p. 355 of our last issue, 1st paragraph, 5th line, delete the word "notwithstanding."

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